TRANSACTIONS

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THE AMERICAN SOCIETY

OF

HEATING AND VENTILATING ENGINEERS

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NEW YORK, JANUARY 19-22, 1915

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THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

TWENTY-FIRST ANNUAL MEETING

New York, January 19th, 20th, 21st and 22nd, 1915.

PROCEEDINGS

Tuesday, January 19th.

Meeting was assembled according to the program at 3 P. M. and owing to the absence of a quorum, was adjourned until Wednesday, January 20th.

Wednesday, January 20th.

FIRST DAY—AFTERNOON SESSION

Meeting called to order by President S. R. Lewis at 1.30 P. M. On motion the calling of the roll and reading of the minutes of the previous meeting was dispensed with.

As the President's address was printed and distributed to the members, it was, at the suggestion of the President voted not to read it before the meeting, so ordered.

President Lewis: We will now ask the Secretary to present his report.

The report was presented by Secretary Blackmore.

On motion the Secretary's report was adopted and ordered to be put in the transactions.

The Secretary then read the report of the Council which asked for a change in the constitution and by-laws.

It was voted that the matter of the changes in the by-laws should be brought up later under the head of new business.

President Lewis: I now call for the report of the Treasurer.

Mr. James A. Donnelly then presented the Treasurer's report, and on motion it was adopted and ordered to be placed in the transactions.

REPORTS OF SECRETARY, TREASURER AND COUNCIL

REPORT OF SECRETARY

The present Secretary took charge of the office March 24th, 1914, the change in officers being due to the former Secretary having to retire owing to ill health.

Volume No. 18 of the Proceedings was mailed to the members during January and February, 1914, and those to the foreign members during the month of March, after the present Secretary took office. Volumes 19 and 20 have been edited and printed during the past year and have now been mailed to the members, thus bringing the transactions of the Society up to date.

Part of one of these volumes has been paid for leaving a balance to be paid on the two volumes of $1,953.46.

No charge for editing these has been incurred this year, the editing having been done in the Secretary's office.

The Secretary has now adjusted the books so that the fiscal and business year of the Society is from January 1 to the close of banking hours on December 31 of any one year in accordance with the resolution to that effect passed by the Society two years ago.

A slight change in the accounts of the previous year was necessary to adjust this as $153.65 was counted in the receipts of 1913 that were really taken in between the first and fifth of January, 1914, and not deposited with the treasurer till that date. Because of this error it was impossible to balance the books between the Secretary and Treasurer. This error has now been corrected and the Secretary's and Treasurer's reports agree.

STATEMENT OF INCOME AND OUTGO

Cash in hand of Secretary, January 2, 1914			$2,823.20	
Less January receipts entered in December, 1913		$153.65		
Less Secretary's suspense account		100.00		
			253.65	
Cash on hand—Treasurer's account—January 2, 1914				$2,569.55
Receipts				
Dues			$4,612.25	
Initiation fees			1,085.00	
Sale of Transactions, net:				
Set vols. 1 to 18	$90.00			
Set vols. 1 to 14	70.00			
On account of complete set	35.00			
Vol. 18	125.25			
Vol. 19	7.50			
Vol. 20	15.00			
			342.75	
Pin badges		98.00		
Sale of Year Book		5.50		
Interest on deposits		47.97		
Sale of electros		37.40		
Sale of reprints		127.00		
Excess exchange		.67		
Telephone call		.05		
Sale of papers		5.90		
			322.49	6,362.49
				$8,932.04
Disbursements				
Transactions:				
Vol. 18, printing		$1,510.77		
Acct. Vol. 19, cuts	$13.32			
Acct. Vol. 19, printing	31.25			
		44.57		
Acct. Vol. 20, cuts	$192.35			
Acct. Vol. 20, printing	392.85			
		585.20		
Postage and express on Vol. 18		105.47		
			$2,246.01	
Pin badges			179.00	
Meetings expense:				
Advance papers		203.85		
Translating foreign papers		15.00		
Stenographer (reporting meetings)		255.00		
Programs		53.40		
Meeting rooms and expenses therewith		171.86		
Registration badges		75.00		
			774.11	
Year Book			180.10	
General administration:				
Salaries		$1,713.00		
Assessments (rent) of office		396.00		
Office stationery and supplies		106.70		
New furniture		73.05		
Storage		39.56		
General printing		338.00		
Ballots		162.03		
Telephone		46.96		
Exchange		7.20		
Postage		215.94		
Bonds of Secretary and Treasurer		10.00		
			3,108.44	
			$6,487.66	
Cash in hands of Secretary (Contingent Acct.)			100.00	
Cash in hands of Treasurer			2,344.38	
				$8,932.04

ASSETS

Furniture and fixtures			$225.00	
Cuts used in printing volumes			250.00	
Stationery and supplies			25.00	
Proceedings				
Vol. 1	134			
Vol. 2	10			
Vol. 3	104			
Vol. 4	124			
Vol. 5	96			
Vol. 6	117			
Vol. 7	106			
Vol. 8	84			
Vol. 9	138			
Vol. 10	128			
Vol. 11	143			
Vol. 12	128			
Vol. 13	135			
Vol. 14	138			
Vol. 15	164			
Vol. 16	161			
Vol. 17	191			
Vol. 18	180			
Vol. 19	181			
Vol. 20	267			
	2,729	at $2.50	6,822.50	
Accounts receivable			45.00	
Library			400.00	
Membership dues outstanding	755			
Less uncollectible (estimated)	100			
			655.00	
Initiation fees outstanding			345.00	
Badges, registration			37.50	
Badges, pin			100.00	
Cash on hand			2,344.38	
				$11,249.38

LIABILITIES

Balance due on Volumes issued to members:		
Vol. 19	$1,178.46	
Vol. 20	775.00	
Membership dues prepaid	47.00	
Accounts payable:		
Storage bill	11.50	
Acct. of postage and express on Vols. 19 and 20 (estimated)	150.00	
		$2,161.96
Membership equity		9,087.42
		$11,249.38

STATUS OF DUES ACCOUNT

Cr.

In arrears, including portion of dues of candidates elected December, 1913	$545.00		
Paying members, January 2, 1914—460 at $10	4,600.00		
Class of Candidates, June 2—20 at $5			
Class of candidates, June 2—3 at $10	137.00		
Class of candidates, June 2—1 at $7			
Class of candidates, Oct. 2—24 at $10			
Class of candidates, Oct. 2—16 at $5	320.00		
Class of candidates, Dec. 2—5 at $10	50.00		
Reinstated members, back dues	80.00		
		$5,732.00	

Dr.

Prepayment dues 1914 account in 1913	$40.50		
Written off, by death	30.00		
Written off, by non-payment of dues	350.00		
Written off, by failure to qualify	20.00		
Total dues received	4,536.50		
		4,977.00	
Dues outstanding			$755.00

STATUS OF INITIATION FEES ACCOUNT

Cr.

In arrears, including portion of candidates elected December, 1913	$235.00		
Class of candidates June 2	340.00		
Class of candidates, Oct. 2	580.00		
Class of candidates, Dec. 2	235.00		
Fee of advancement from junior grade	5.00		
		$1,395.00	

Dr.

Total initiation fees received	$1,020.00		
Written off, by failure to qualify	30.00		
		1,050.00	
Total fees outstanding			$345.00

Recapitulation

Total debit accounts of members	$1,100.00

During the month of December storage space was secured in the United Engineering Societies' Building, to which all the volumes of transactions and papers that have been in storage for some years have been removed.

There were two reasons for doing this. One was that additional storage space would be necessary to store volumes 19 and 20, which would have cost more than the new space is costing. The other was that the facility for obtaining insurance on the property is better by having all in one place.

The cost of operating the Society per capita this year is $11.94. The increased cost of the transactions this year and the printing of a year book has to some extent caused this increase, which was predicted in the Secretary's report last year.

Year	Total Receipts		Total Expenses	Mean number of members	Expense per member
1907	$3,912.41		$3,662.45	301	$12.20
1908	3,928.93		4,025.84	335	12.00
1909	3,952.14		4,687.41	357	13.13
1910	4,324.10		4,269.83	377	11.36
1911	4,269.83		4,793.08	405	11.83
1912	4,717.40		5,013.19	437	11.47
‡1913	6,160.14		4,400.90	455	9.67
1914	6,362.49	$6,587.66			
* Less extra on transactions		629.67			
† Less papers and electros sold		164.40			
		———	$5,793.59	485	11.94

* This amount was paid out in addition to a full year's transactions, see statement.

‡ No year book issued and the small amount of printing done and the small size of the transactions made this year's expense lower than usual.

† This item represents commercial transactions, cost of which is charged to the expenses of the year, hence the deduction.

During the past year, at least so far as the semi-annual meeting was concerned, all the papers were printed that were presented at the meeting, whereas in previous years a number was presented in manuscript. This makes quite a difference in the size of the printing bills, but such printing is very necessary for the good working of the Society.

During the past year a Brochure was published which sets forth some of the activities of the Society and with this and other printed matter an active campaign for new members will be carried on during 1915 if the members will co-operate with the administration.

The Secretary has learned that the officials of states and cities are willing to listen to the recommendations of societies provided that such societies are strong enough to represent the industry or profession they stand for.

Our Society is not strong enough, with its present membership, to command the respect it deserves from such interests, but it would command it, if its ranks could be increased up to or over one thousand members. There is plenty of eligible material in the industry from which to choose twice that number.

The Secretary much appreciates the assistance given by a number of the members in securing new candidates for membership in the Society, but there is yet a great deal of work that can be done to get new members and more co-operation on the part of all the members is earnestly desired to spread the influence of the Society.

The Secretary desires to express his appreciation to the Cleveland Entertainment Committee, who undertook the charge of the Cleveland convention last summer, and largely to them is due the credit for a very successful summer meeting.

Respectfully submitted,

J. J. BLACKMORE, Secretary.

January 21, 1915.

REPORT OF TREASURER

The Treasurer makes the following report for the fiscal year of 1914, from January 1st to December 31st:

	RECEIPTS		EXPENDITURES		
January:					
Balance		$2,569.55			
From Secretary	$534.10				
Interest	3.39		Exchange	$ .20	
		537.49			$ .20
February:					
From Secretary	$1,688.74				
Cr. exchange	.02		Vouchers 127-147	$1,191.30	
Interest	3.39		Exchange	.27	
		1,692.15			1,191.57
March:					
From Secretary	$538.60				
Cr. exchange	.06		Vouchers 148-150	$165.96	
Interest	4.60		Exchange	1.26	
		543.26			167.22
April:					
From Secretary	$531.75				
Cr. exchange	.16		Vouchers 151-158	$1,770.90	
Interest	5.09		Exchange	.61	
		537.00			1,771.51
May:					
From Secretary	$165.00				
Cr. exchange	.06		Vouchers 159-165	$318.02	
Interest	4.93		Exchange	.70	
		169.99			318.72
June:					
From Secretary	$780.00		Vouchers 166-170	$167.42	
Interest	3.67		Exchange	.20	
		783.67			167.62
July:					
From Secretary	$143.60		Vouchers 171-179	$1,218.78	
Interest	4.27		Exchange	1.21	
		147.87			1,219.99

RECEIPTS—Continued			EXPENDITURES—Continued		
August:					
From Secretary ...	$288.60				
Cr. exchange	.14		Vouchers 180-184 ...	$342.83	
Interest	4.71		Exchange	.40	
		293.45			343.23
September:					
From Secretary ...	$103.80		Vouchers 185-187 ...	$90.35	
Interest	4.05		Exchange	.52	
		107.85			90.87
October:					
From Secretary ...	$658.10		Vouchers 188-194 ...	$366.49	
Interest	3.29		Exchange	.40	
		661.39			366.89
November:					
From Secretary ...	$232.30		Vouchers 195-204 ...	$455.53	
Interest	3.29		Exchange	.83	
		235.59			456.36
December, 1914:					
From Secretary ...	$649.35				
Cr. exchange	.14		Vouchers 205-212 ...	$493.08	
Interest	3.29		Exchange	.40	
		652.78			493.48
		$6,362.49			$6,587.66

SUMMARY

From Secretary	$6,313.94				
Cr. exchange	.58		Vouchers	$6,580.66	
Interest	47.97		Exchange	7.00	
		$6,362.49			$6.587.66
Balance Jan. 1, 1914		2,569.55	Balance Jan. 1. 1915		2,344.38
		$8,932.04			$8,932.04

Respectfully submitted,

JAMES A. DONNELLY, Treasurer.

REPORT OF COUNCIL

The Council makes the following report of its administration of the affairs of the Society since the Council organized on January 21, 1914, immediately following the last annual meeting.

The Council has maintained the Society's headquarters in the Engineering Societies' Building, 29 West 39th Street, New York City, and the office of the Secretary has been frequently used by individual members and by committees for other metings. The Council has held seven, and the Executive Committee eleven meetings during the year. Several members who could not attend certain meetings contributed their views in writing in advance on subjects to be considered at those particular meetings.

Owing to the resignation of Secretary E. A. Scott soon after the last annual meeting, the Council had to secure a new Secretary, and

the Council concluded that a secretary was needed who could give more time to the work than was possible for previous secretaries to give to the Society. The members of the Council agreed that the Society would progress more rapidly if a secretary could be kept constantly employed in its interest.

The Council prevailed upon Mr. J. J. Blackmore to accept the office as stated in the circular to the members of April 15, signed by the Executive Committee. The statements of the year show the value of the appointment made, as the membership has grown during the year to a greater extent than ever before and the income has materially increased.

The Council feels that the prosperity of the Society and its influence for good will be greatly enhanced if a permanent secretary can be retained, who will devote his whole time during business hours to its affairs. The Council also believes that the Secretary of the Society should be conversant with every branch of engineering that relates to heating and ventilation, for often the Secretary has to represent the Society before engineering schools and societies to secure co-operation in the work of the Society.

The summer meeting of the Society was held at Cleveland on July 9, 10 and 11 and was notable in that the attendance was greater than at previous summer meetings. A valuable opportunity was also afforded the members at that meeting to visit the Park and buildings of the National Lamp Works. During the visit the members and guests listened to an instructive lecture on light and were otherwise entertained in an agreeable manner.

The Society has been represented at meetings during the year as follows: At the joint meeting of the National Association of Master Steam and Hot Water Fitters, the American Society of Mechanical Engineers, and other societies in Washington for the standardizing of flanges, etc., by Mr. N. S. Thompson and Mr. C. R. Bradbury. At the annual meeting of the Society for the Promotion of Engineering Education at Princeton, N. J., by the Secretary, and at several meetings of the American Society of Mechanical Engineers' Special Committee, which was appointed to formulate uniform standards for all kinds of steam boilers and pressure vessels, by P. H. Seward, U. G. Scollay and J. J. Blackmore.

A statement of the assets and liabilities of the Society is given in an accompanying table. It is interesting to note that the cash on hand is in excess of the obligations of the Society and that the affairs of the Society have been so conducted since its organization that the equity of each member is in excess of his original initiation

fee. In other words, on the basis of 507 members the equity per member on December 31, 1914, was $17.92.

The present administration has, by suggestion and otherwise, started many new lines of investigation which it is hoped will lead to a solution of the unsolved problems related to the art of heating and ventilation.

The details in the changes of membership are shown in the accompanying table. Three ballots were convassed in the usual manner: one on June 2, one on October 2 and another on December 19, adding 80 members to the rolls. This increase in membership, however, has been partially offset by losses from death, resignation and non-payment of dues. Two candidates reported one year ago as being elected on the ballot of December 19, 1913, failed to qualify.

The membership has increased by 45 during the year and now numbers 507.

STATUS OF MEMBERSHIP

Honorary members:				
Total number, January 2, 1915				2
Members:				
Total number, January 20, 1914			402	
Accessions by election	56			
by reinstatement	1			
by advancement from associate	2			
by advancement from junior	1			
		60		
Losses by resignation	8			
by non-payment of dues	13			
by death	6			
		27		
Net increase			33	
Total number January 2, 1915				435
Associate members:				
Total number, January 20, 1914			42	
Accessions by election		15		
Losses by resignation	5			
by non-payment of dues	2			
by advancement to full	2			
		9		
Net increase			6	
Total number, January 2, 1915				48
Junior members:				
Total number, January 20, 1914			16	
Accession by election		9		
Loss by resignation	1			
by non-payment of dues	1			
by advancement to full	1			
		3		
Net increase			6	
Total number January 2, 1915				22
Total membership January 2, 1915				507
Total membership January 20, 1914				462
Total net increase				45

The Society suffered the loss of six members by death during the year. The list includes a past vice-president, A. B. Franklin, who died August 22. The other members were: H. W. E. Muellenbach, April 17; W. A. Gates, May 24; F. L. Busey, June 7; E. B. Denny, July 12, and H. A. Vollbracht, December 23.

Appropriate action has been taken by the Council to convey the condolence of the Society to the families of the bereaved.

Those who have honorably withdrawn from membership by resignation were: C. H. Basshor, Cambridge, Md.; C. H. Davis, New York City; H. C. Faulkner, Ogdenburg, N. J.; Walter Jones, The Uplands, Strourbridge, Eng.; L. H. Prentice, Chicago, Ill.; H. H. Rosenbaum, St. Louis, Mo.; W. C. Vrooman, Schenectady, N Y.; F. A. Williams, New York City; W. A. Birdsall, Newark, N. J.; Chas. F. Chase, New York City; G. L. Greenman, New York City; M. E. Monash, New York City; Frank Schreidt, Mansfield, Ohio; W. Mayer, Jr., W. New York, N. J.

CONSTITUTION AND BY-LAWS

In the working of the constitution and by-laws as amended last spring some additions have been found desirable.

Article III, section 2, should be amended to enable the Society to admit as members gentlemen of the learned professions, and factory, health and municipal inspectors who may be working on heating problems or inspecting heating and ventilating plants. The last six lines should be changed to read as follows:

"Mining, civil, electrical, mechanical, naval or Government engineers, chemists, physicians, health or ventilation inspectors, scientists or architects who are, in the opinion of the Council, qualified by reason of their experience in designing, improving, inspecting, investigating or developing the art of heating or ventilation, may also become members."

It is also desirable that means should be provided for the endorsing of applicants who may want to join the Society from places in which we have no membership. It is believed that endorsements as to the standing of such an applicant from local trade or engineering societies, or from the officers or members of faculties of educational institutions, will be quite as advantageous in determining the qualifications of a candidate for admission to the Society, as the endorsement of Society members. The American Society of Mechanical Engineers, with over 6,000 members has found such a provision necessary.

It is, therefore, recommended that Article IV, section 1, should be changed to read as follows:

"In case applicants for membership are not acquainted with members of the Society and there are no local members in their vicinity, references or endorsements may be considered from officers or members of faculties of educational institutions, or from the officers of local engineering or trade societies in good standing in the community, or from manufacturing firms or corporations who are well acquainted with the applicant."

Article V, in practice, increases the work of bookkeeping and makes the annual statement difficult to adjust. It is found to be better that all dues should be payable each year on January 1.

It is, therefore, recommended that Article V be changed beginning with the sentence on the eleventh line as follows:

"The annual dues of members elected during the year may be pro-rated for the balance of said year, but such candidate paying less than five dollars for the pro-rated time shall not be entitled to receive the volume of transactions for the year in which he is elected. Upon the payment of the initiation fee and his pro-rata annual dues for the first year, the person elected shall be entitled to the rights and privileges of membership in the grade to which he was elected."

SAMUEL R. LEWIS, Chairman.

President Lewis: We will now hear the report of the Committee appointed to prepare a set of minimum ventilating requirements for public and semi-public buildings.

In the absence of Prof. Hoffman extracts from the paper were read by Mr. F. T. Chapman, a member of the Committee.

President Lewis: This Committee has had several meetings which the members attended at their own expense and as they were widely separated the expense must have been a considerable item. The report therefore merits and requires our most careful consideration.

It was moved by Mr. F. K. Chew and seconded by Secretary Blackmore that the report be accepted subject to some slight changes that may be necessary, and the committee to have charge of it until it is completed.

Considerable discussion of the various recommendations obtained. The speakers being Mr. M. W. Franklin, Mr. S. R. Lewis, Mr. J. D. Cassell, Mr. F. W. Lamb, Mr. J. S. Otis, Mr. F. T. Chapman, Mr. F. K. Davis, Mr. H. M. Hart, Mr. Johnson, Mr. D. D. Kimball and Mr. F. K. Chew, after which the motion was changed as

follows: That the report of the committee be adopted subject to the approval of the Council, after the committee had agreed upon the changes that were necessary to be made in the report to meet the objections raised by the discussion. Carried unanimously by standing vote.

President Lewis: The Secretary will now present the paper, Development in Heating and Ventilating Industrial Buildings by E. L. Hogan.

President Lewis: The paper is ready for discussion.

The paper was discussed by Mr. F. K. Chew, Mr. M. W. Franklin, Mr. S. R. Lewis, and Mr. Bolton and it was voted to refer it to the committee on standards.

President Lewis: A Report of the Committee on the Best Place to Locate Radiators in Rooms from W. F. Verner will be read by the Secretary, Mr. J. J. Blackmore.

It was a report of progress only, referred back to the committee for report at the semi-annual meeting.

We will now proceed to the appointment of tellers to count the ballots for the officers of the Society for the ensuing year, who will report at the evening session.

Committee appointed: Mr. F. K. Chew, Chairman; Mr. F. K. Davis, Mr. C. A. Blaney.

President Lewis: We will now proceed with the election of a Nominating Committee for officers and Council for the year 1916.

According to the constitution and by-laws, it is necessary for a nominating committee of five to be chosen at this meeting by ballot. I appoint Mr. R. E. Lynd, Mr. A. S. Armagnac and Mr. J. S. Otis to act as tellers.

Nominations were made and the tellers were asked to report the results later in the session.

President Lewis announced the fact that Mr. W. J. Baldwin had been recommended to the council for honorary membership and that the Council had acted favorably on the application. Mr. Baldwin's name is now before you to vote for his election as Honorary member of this Society. The motion being put, Mr. Baldwin was elected unanimously.

Mr. Baldwin was then presented to the Society by Past-President John F. Hale and President Lewis.

Mr. Baldwin expressed his appreciation in a suitable manner.

President Lewis: The members of our Society in St. Louis have organized a Chapter for St. Louis and vicinity. They have secured permission of the Council and now come before us requesting our sanction to their incorporation as a Chapter.

It was moved and seconded that the Society grant a charter for a Chapter to be located at St. Louis and it was so voted unanimously.

President Lewis: We will now discuss the amendments recommended by the report of the Council, to the Constitution. In order to facilitate matters, I will ask the Secretary to read the proposed amendments.

The amendments were read by the Secretary.

The proposed change to Article III, Section 2, occasioned considerable discussion by Mr. J. I. Lyle, Mr. M. W. Franklin, Mr. F. K. Davis, Mr. W. J. Baldwin, Mr. F. K. Chew, Mr. D. D. Kimball, Mr. Perry West, Mr. D. M. Quay, Mr. John F. Hale, Secretary Blackmore, Mr. H. J. Barron, Mr. H. M. Hart, Mr. J. A. Donnelly and it was voted to make no change in Article III, Section 2.

As to changes in Article IV, Section 1, and Article V, it was voted unanimously to submit a ballot to the members for their approval of the changes in these two sections as recommended in the Council report.

The report of the Tellers Committee appointed to count the ballots for the election of a committee on nominations gave the following names as duly elected for that purpose:

Frank K. Chew, W. G. Snow, J. I. Lyle, Bert C. Davis and H. J. Barron.

President Lewis: I therefore declare these gentlemen a committee for nominating officers for the election of 1916.

FIRST DAY—EVENING SESSION

Wednesday, January 20, 1915.

Meeting was called to order by President Lewis at 7.30 P. M. who asked for the Report of the Tellers of Election for Officers for 1915-16. Mr. F. K. Chew, chairman.

Mr. Chew: The Tellers found a total of 157 votes cast, three of which were rejected. Two of them came in plain envelopes with no name on the outside designating the member voting. Another ballot was rejected because other names were voted for without scratching off the names of the nominees.

Ballots figuring in the election total, 154.

For President, D. D. Kimball	148
For Vice-President, H. M. Hart	149
For Second Vice-President, F. T. Chapman	139
For Treasurer, Homer Addams	154

FOR COUNCIL

F. I. Cooper	149	J. T. J. Mellon	142
W. M. Kingsbury	146	H. C. Meyer, Jr	141
S. R. Lewis	146	A. K. Ohmes	141
F. G. McCann	142	Dr. E. V. Hill	140

Respectfully submitted,

F. K. CHEW,
F. K. DAVIS,
C. A. BLANEY,

President Lewis: According to the Constitution and By-Laws, I hereby declare these officers duly elected for the usual term.

President Lewis: We will now finish that part of the program which was to have been completed this afternoon, namely, the paper on An Experiment With Ozone as an Adjunct to Artificial Ventilation at the Mt. Sinai Hospital, New York City. By Mr. A. M. Feldman.

Paper read by Mr. Feldman.

President Lewis: The paper is before you for discussion.

The paper was discussed by Mr. Thos. Barwick, Mr. M. W. Franklin and Mr. A. M. Feldman.

President Lewis: The next paper, "The Centrifugal Fan," by the late Mr. F. L. Busey, was written by him some time previous to his death and it was sent to the Society through the kindness of Mrs Busey. Mr. Carrier was closely associated with Mr. Busey and I will ask him to present the paper.

Mr. W. H. Carrier: I was very closely associated with Mr. Busey and I am able to testify as to the thorough interest with which all his work was conducted and the great effort he always put into everything he brought before the Society. While this paper is somewhat of a historical nature, I know personally that many of the data in regard to two or three of the types of fans mentioned here were prepared by him and that makes it of especial interest, as the work given is, on that account, authoritative.

Mr. A. M. Feldman: I move that the Society send thanks to Mrs. Busey for the paper.

President Lewis: Moved and seconded that a rising vote of thanks be given Mrs. Busey and that the Secretary be instructed to transmit such a letter to Mrs. Busey. So voted unanimously.

President Lewis: We will now consider the paper—"Engine Condensation" by Mr. Perry West.

Mr. Perry West: With permission of the President and members I shall abbreviate the reading of the paper to give more time to the discussion.

The paper was discussed by Mr. F. R. Ellis, Prof. Wm. Kent, Mr. David Moffat Myers, Mr. J. S. Otis, Mr. W. H. Carrier, Mr. C. M. Ripley and Mr. J. A. Donnelly.

President Lewis: Inasmuch as we have a paper "Heating Value of Exhaust Steam," by Mr. David Moffat Myers, that treats of the same subject, we will hear Mr. Myers' paper and then continue the discussion.

Mr. Myers then read his paper.

The discussion was continued by Prof. Kent, Mr. W. H. Carrier, Mr. C. M. Ripley, Mr. J. A. Donnelly, Mr. D. M. Quay, President Lewis and Secretary Blackmore.

The meeting adjourned at 11:00 P. M.

MORNING SESSION—THURSDAY, JANUARY 21, 1915.

Meeting called to order by President Lewis at 10:30 A. M. and the session started at once with the paper—"A Description of the Experimental Plant of the N. Y. State Commission on Ventilation, with the Record of the Results of a Number of Experiments that have been made." D. D. Kimball and Geo. T. Palmer.

The paper was presented by Mr. D. D. Kimball.

President Lewis: The paper is before you for discussion.

The paper was discussed by Mr. Hart, Prof. Winslow, J. H. Davis, President Lewis, F. K. Davis and Mr. Chew.

President Lewis: We will now proceed to the next paper, "Studies in Air Cleanliness," by Profs. M. C. and G. C. Whipple.

In the absence of the authors the paper was read by Secretary Blackmore, and it was discussed in connection with papers of Messrs. Bolton and Grady.

President Lewis: We will now proceed with the paper, "The Problem of City Dust," by R. P. Bolton.

Mr. Bolton read the paper.

It was discussed by Mr. D. M. Quay and President Lewis.

President Lewis: We will now listen to the paper, "Cinder Removal from Flue Gases of Power Plants," by C. B. Grady.

The paper was discussed by Mr. R. P. Bolton, Mr. W. J. Baldwin, Mr. W. H. Carrier, Mr. F. T. Chapman, Mr. H. M. Hart, Mr. J. H. Davis and President Lewis.

President Lewis: The Society wishes to extend to Messrs. Grady and Whipple, its thanks for their kindness in preparing these papers.

On motion the meeting was adjourned at 12:30 P. M.

SECOND DAY—AFTERNOON SESSION

Thursday, January 21*st,* 1915.

Meeting was called to order by President Lewis 2:10 P. M.

Paper—"A Study of Heating and Ventilating Conditions in a large Office Building." Prof. C. E. A. Winslow.

Paper was read by Prof. Winslow.

President Lewis: On behalf of the Society I wish to thank Prof. Winslow for preparing this paper.

The paper was discussed by Mr. H. M. Hart, Mr. D. D. Kimball, President Lewis, Prof. C. E. A. Winslow, Mr. J. H. Davis, Mr. F. K. Davis and Mr. F. K. Chew.

President Lewis: We will now proceed with the next paper, "Recirculating Air in a Schoolroom in Minneapolis," by Prof. Frederic Bass.

Professor Bass not being present the paper was read by Secretary Blackmore.

The paper was discussed by Prof. Shepherd, Mr. W. H. Carrier, Mr. A. K. Ohmes, President Lewis, Mr. F. K. Davis, Mr. H. M. Hart and Mr. F. G. McCann.

President Lewis: We will now present the paper, "The Ventilation of Sleeping Cars; Comparative Tests of Exhaust Ventilators," by Dr. T. R. Crowder.

This paper was discussed by Mr. H. M. Hart, President Lewis, Mr. F. K. Davis, Mr. J. F. Hale, Mr. Riley, Mr. F. G. McCann, Mr. D. M. Quay and the various questions asked were replied to by Dr. Crowder.

President Lewis: We will now present the paper, "Ventilation of Industrial Plants." C. T. Graham-Rogers, M. D. and W. T. Doyle, M. E.

The authors not being present the paper was read by the Secretary.

The paper was discussed by Mr. A. M. Feldman, Mr. H. M. Hart, President Lewis, Mr. F. K. Davis, Secretary Blackmore and Mr. W. J. Baldwin.

President Lewis: One of the papers that was to have been presented last night will now be offered for discussion, "Test of a Sectional Downdraft Boiler," by Mr. C. A. Fuller.

The paper was read by Mr. Fuller.

The paper was discussed by Mr. A. M. Feldman, Mr. W. J. Baldwin, Mr. H. M. Hart, Mr. J. D. Cassell, Mr. D. M. Quay, Mr. F. C. Bartley, Mr. Davis, Mr. J. S. Otis, Mr. C. R. Bradbury, Mr. S. M. Bushnell, Mr. Homer Addams, Mr. G. W. Martin, Mr. F. R. Ellis, and the several questions raised were replied to by Mr. Fuller.

On motion the meeting adjourned at 5:30 P. M.

THIRD DAY—AFTERNOON SESSION

The meeting was called to order by President Lewis at 2 P. M., and opened with a paper "The Threading of Steel vs. Wrought Iron Pipe," by Clifford G. Dunnells.

Owing to the absence of Mr. Dunnells, the paper was presented by Mr. Frank N. Speller, who made several explanations as to the manner of making the tests, which will be printed with the paper.

President Lewis: We will now present the paper, "Some Phases of Room Heating by means of Gas Appliances," by George S. Barrows.

The paper was read by Mr. Barrows and was discussed by Mr. C. A. Blaney, Mr. H. J. Barron, Mr. F. T. Chapman, Prof. Wm. Kent, Mr. H. M. Hart, Mr. F. K. Davis, Mr. M. W. Franklin and Mr. G. T. Palmer, after which the various questions were answered by Mr. Barrows.

President Lewis: We will now proceed with the paper, "The Capacity of Steam Pipes at Different Pressures," by J. S. Otis.

Mr. Otis read the paper by extracts.

It was then discussed by Prof. Wm. Kent, Mr. J. A. Donnelly, Mr. H. J. Barron, Mr. M. W. Franklin, President Lewis, and Mr. Otis replied to the several questions raised.

President Lewis: The paper and the discussions will be referred to the Publication Committee and the Council, to see whether corrections can be made in the paper to make it acceptable for the transactions of the Society.

President Lewis: We will now present the paper, "Rational Methods applied to the Design of Warm Air Heating Systems," by Roy E. Lynd.

Mr. Lynd read only extracts from the paper and it was discussed by Mr. D. M. Quay, Prof. Wm. Kent, Mr. Mobley, and a written

discussion was read from Mr. F. W. Colbert, after which Mr. Lynd replied to the several questions raised by the discussion.

President Lewis: In departing from the Chair as President of the American Society of Heating and Ventilating Engineers, I wish to acknowledge the very great help given by our Secretary. The Secretary has given his entire time for a very small part of what it is worth and to whom a very large part of the success of the administration this year is due. I call for a rising vote of thanks for his endeavors on behalf of the Society.

Carried unanimously.

Mr. Blackmore: Mr. Chairman and Gentlemen: Without attempting to make a formal speech, I thank you for your expression of approval of my work, and I will endeavor to deserve your further appreciation if I should again be appointed to the office.

President Lewis: I wish to acknowledge the very great help I have received from the members of the Council, particularly from Messrs. Chapman, Macon, Kimball, and Capron. Others have done their part and have done it well, but these gentlemen have been particularly helpful to me in the work of my office.

President Lewis: We will now proceed with the installation of the newly elected officers.

Eighteen years ago Mr. W. M. Mackay was president of this Society and it affords me great pleasure to yield the chair to Past-President Mackay that he may install the officers.

Past-President Mackay then took the chair, and made a speech appropriate to the occasion.

Mr. Wm. M. Mackay: In installing those you have elected to serve as your officers, I would request the assistance of two Charter members, Mr. H. J. Barron of New York and Mr. B. H. Carpenter of Wilkes-Barre. If these gentlemen will kindly present such of the elected officers as are present, I will be pleased to install them commencing with President-elect, D. D. Kimball.

Mr. Kimball was then escorted to the chair.

Mr. W. M. Mackay: Mr. Kimball we greet you as President of this Society. We have had 21 presidents during our growth from infancy to manhood and you, as the 22nd president, will stand as the first president of a society which is now fully matured and responsible for all its acts.

Mr. W. M. Mackay: First Vice-President, H. M. Hart of Chicago and Second Vice-President, F. T. Chapman of New York, will now be escorted to the Chair.

Mr. W. M. Mackay: Mr. Hart, I greet you as First Vice-President of this Society. Mr. Chapman, we have been associated

together a long time. You have done excellent work for the New York Chapter, as we all know, and there could be no better choice for the office of Second Vice-President.

Mr. Addams was then escorted to the Chair.

Mr. Mackay: Mr. Addams, I congratulate the Society on having elected you as its Treasurer. You are one of those who have always been with us and gave us help when help was needed. I take great pleasure in presenting you to the Society as the duly elected Treasurer.

Mr. F. I. Cooper, Mr. W. M. Kingsbury and Mr. F. G. McCann being the only members of the newly elected Council present, were escorted to the Chair and duly installed.

Mr. Mackay then in a few well chosen remarks in which he said the newly elected officers would be expected to make speeches, vacated the Chair in favor of the newly elected President, Mr. D. D. Kimball.

President D. D. Kimball: "Mr. President, Ladies and Gentlemen, I am not at all unmindful, as I stated last night at our annual dinner, of the responsibilities imposed by this office. To those of you who have not been so closely associated with the work, there may not be a full appreciation of the responsibility involved. The work is not easy, nor light. I realize that to a large extent rests on my shoulders the maintenance of the progress of the Society. Indeed, I would be glad if we could improve on the past President's diagram to the extent of making these lines divert to a greater angle, and that our Society might grow at a more rapid rate.

This responsibility is indicated by the increased membership we have had during recent years, by the increasing influence which the Society has, and by the increasing number and improved character of the papers. However, all of this increase in interest and work could not go on if the responsibility rested on the shoulders of any one man, much less on mine. So that to bring about these results I shall expect, and it shall be my aim, to spread this responsibility over the shoulders of the other members of the Society, over the shoulders of the members of the Committees possibly, but not that alone, I shall hope and expect that you will all bear a portion of this responsibility, that you will evidence your interest in the work by willingness to provide papers and suggestions and even criticisms, for they, perhaps, are quite as helpful as anything.

It is needless to say that the greatest dependence may be placed upon the Council for the incoming year, and knowing these gentlemen very well, for with several of them I have been associated for some time, I am not afraid of placing dependence upon them.

I shall especially be dependent upon the Secretary of the Society, and having worked with him through the period of last year during which he has been our Secretary, I know I may count with assurance upon him. Similarly, to no small extent is dependence placed upon the assistant to the Secretary.

I need say very little regarding the program which I have in mind, for I spoke at length about that last evening.

I think possibly that I should, and I feel that I must, voice the feelings of the members of the Society towards their retiring officers. It is a bit of a handicap to work at widely separated points, but that certainly has been more than overcome by the energy and faithfulness of our retiring President and fellow members of the Council, so that to each of them I want to express for the Society our appreciation and thanks, and most especially so to Mr. Lewis.

Reference has been made to the Old Guard. For my part, I see no excuse or apologies necessary for the Old Guard. They did splendid work and all that we do to-day and all for which we hope in the future is being built, and can be continued, on the foundation which they laid for us.

I thank you for the honor and I pledge you the best of my services in the interests of the Society.

It has been stated that the other officers elected should speak to you. I will first present Mr. Hart, our Vice-President.

Our First Vice-President is modest, but he has been one of those Chicago members who has been a real worker and a source of very great help and encouragement to us in our work.

The Second Vice-President has been associated in the work for some years. He is one of those men in whose hands a matter can at any time be placed with assurance that it will be attended to. He is, indeed, one of the busiest and best of workers. I also present Mr. Chapman."

Mr. H. M. Hart: "Fellow Members, I don't think I can add much for the good of the Society to what our President has said. I wish to say that I certainly appreciate the honor this Society has bestowed upon me in electing me as Vice-President.

I have not felt and don't feel that I am worthy of the honor and I don't feel that I have done enough for the Society to warrant them bestowing that honor upon me at this time and the only way I can square myself with the members will be to put my shoulder to the wheel and try and justify their action by increased energy on my part for the advancement of the Society.

I am one of those who are being educated in this Society. I am getting a great deal of knowledge from these meetings and while

I am on that theme I might add, inasmuch as the ladies have been kind enough to come here, my appreciation of what I gained in the meetings they have attended. I have accomplished something at each banquet; this time I learned to do the fox trot which I did not know before, hence I feel that my whole education is progressing as well as can be expected.

Of course, we need papers and we need good ones and I wish that everybody might appreciate the privilege of being on a Committee to prepare a paper to present to this Society. It gives them opportunities which they might never otherwise grasp to learn something. It makes them dig up things they would never otherwise find and my experience has been that committee work is of greater benefit to the members of the Committee than it is to the Society. They gain knowledge they never would otherwise gain. The attendance at the meetings is most important. The papers are distributed to all the members, but I find that the discussion here in open meetings is really the valuable part of the proceedings and I for one feel that I gain much by each meeting that I attend.

I hope that I will be able to do justice to the office although I don't feel that I will have much responsibility as Vice-President, but as a member of the Council, I do feel a responsibility. I thank you for electing me.

Mr. Chapman: Brother Members and Ladies, I take pleasure in congratulating you upon the progress our Society is making and upon the increasing interest shown the past year in using opportunities for service. I feel strongly that seeking out and using opportunity for service is one of the most important things for us to carry in mind and to cultivate as a definite policy.

I am deeply interested in the Society and am very glad to continue to co-operate with you in working for its welfare.

Mr. Homer Addams: I thank you heartily for the expression of confidence you have in me in supporting me for this position, and am sure that fellowship in this work is not a burden, for those of us who linger and labor here together. I hope you will have no regrets for having supported this ticket and that you will keep the treasurer promptly supplied with funds as required by your obligations to the Society.

Mr. D. D. Kimball: The last suggestion is most pertinent. It becomes necessary that there shall be a meeting of the Council immediately because of the fact that some members must leave the city shortly, therefore I will ask the members of the Council if they will adjourn to the office of the Society on the seventh floor

and will ask Past-President Mackay to assume the Chair for the remainder of this session.

Council then adjourned to meet in special session.

Past-President then took the chair.

S. R. Lewis: I wish to propose a vote of thanks to the Entertainment Committee of the New York Chapter, also the New York ladies who have assisted.

. Past-President Mackay: Moved and seconded that a vote of thanks be extended to the Entertainment Committee of the New York Chapter and to the ladies who assisted them in entertaining the guests. Those in favor, please rise. A unanimous vote.

Past-President Mackay: Any further discussion on the paper on Hot Air Heating? If not the next paper is on "Crude Oil as Fuel in Low Pressure Heating Systems." Mr. H. S. Haley.

Mr. Haley is not present and as the paper has been in the hands of all we will refrain from reading it, but if any of the members have any discussion on it, shall be glad to hear it at this time. No discussion.

Any new business? If not, a motion to adjourn will be in order.

It was moved and seconded that the meeting adjourn. So voted Past President Mackay declared the meeting adjourned. Sine Die.

LIST OF MEMBERS AND GUESTS PRESENT AT THE TWENTY-FIRST ANNUAL MEETING, JANUARY, 1915

New York, January 22, 1915

MEMBERS

Addams, Homer
Armagnac, A. S.
Alt, H. L.
Barron, H. J.
Baldwin, W. J.
Barwick, Thos.
Bendure, J. A.
Blackmore, J. J.
Blackman, A. O.
Blaney, C. A.
Bolton, R. P.
Bradbury, C. R.
Bradley, J. T.
Boyden, D. S.
Callahan, M. J.
Carpenter, B. H.
Cassell, J. D.
Carrier, W. H.
Chapman, F. T.
Chapman, D. W.
Chew, F. K.
Claffey, E. J.
Cooper, F. I.
Curry, H. W.
Cyphers, J. F.
Davis, F. K.
Davis, J. H.
Davis, Bert C.
Doherty, P. C.
Donnelly, J. A.
Dewey, W. H.
Driscoll, W. H.
Eadie, J. G.
Edgar, A. C.
Feldman, A. M.
Fabricus, P. H.
Farnham, G. D.
Febrey, E. J.
Flett, H. R.
Foster, W. M.
Franklin, M. W.
Fleisher, W. L.
Fuller, C. A.
Gardner, S. F.
Gates, H. T.
Geiser, H.
Greey, Geo. V.
Greene, H. R.
Goodnow, W. F.
Gordon, H. B.
Gombers, H. B.
Graham, Jos.
Grimshaw, G. E.
Hale, John F.
Hanbury, John F.
Hart, H. M.
Heath, F. R.
Hunt, R. B.
Issertell, H. G.
Jellett, S. A.
Kimball, D. D.
Kingsbury, W. M.
Kent, Wm.
Kiewitz, Conway
Kiewitz, A. A.
Kline, W. J.
Koithan, W. S.
Knight, Geo. W.
Lamson, F. S.
Lennox, F. J.
Lewis, S. R.
Lemmey, Robt.
Linn, Homer R.
LeCompt, W. G.
Lyle, J. I.
Lynd, Roy E.
Macon, W. W.
Mackay, W. M.
Mallory, H. C.
McCann, F. G.
McIntire, J. F.
Mobley, E. S.
Morrison, Chas.
Murphy, W. R.
Mellon, J. T. J.
McKiever, W. H.
Myers, D. M.
Myrick, J. W. H.
Merritt, J. H.
Meyer, H. C., Jr.
Nelson, Benj.
Mandeville, E. W.
Otis, James S.
O'Hanlon, G.
Olvany, W. J.
Parter, S. C.
Petry, C. W.
Pearce, C. E.
Phegley, Frank G.
Purcell, A. J.
Pryor, F. S.
Quay, D. M.
Ritter, Arthur
Riley, C. L.
Robertson, G. A.
Ritchie, Wm.
Staten, C. H.
Scott, C. E.
Schmidt, G. G.
Seward, P. H.
Smallman, W. T.
Snow, W. G.
Stockwell, W. R.
Strader, B. K.
Schloss, N. L.
Stone, E. R.
Speller, F. N.
Sherman, L. B.
Timmis, W. S.
Townsend, A. E.
Treat, E. J.
Teran, C.
Turno, W. G.
Underhill, W. W.
West, Perry
Welsh, H. S.
Warsop, C. E.
Wilson, F. A.
Webster, Warren

GUESTS

Chas. Ackerman
Geo. W. Backoff
W. E. Barnes
H. C. Beatty
E. J. Bender
S. C. Bloom
A. D. Brady
J. A. Brokow
A. P. Broomell
W. S. Buchanan
C. A. Bulkeley
H. C. Carson
Paul Chapman
A. Creekmore
T. R. Crowder
R. C. Doremus
L. L. Driggs
G. A. Eagan
E. B. Ellison
H. J. Engle
W. H. Ennis
J. B. Garfield
O. E. Goldschmidt
R. K. Goodlatte
Chas. B. Grady
A. D. Granger
E. Grassler
H. E. Hart, Jr.
J. M. Hasalson
Aug. Heimann
Jos. Herzstein
J. C. Hutchinson
C. C. Johnston
R. C. Jones
H. L. Jones
F. W. S. King
F. T. Kitchen
W. J. Knox
C. E. Krausse
E. K. Lanning
C. E. Laurence
L. L. Lewis
Henry Lipkeman
E. J. K. Mason
Robt. Mayor, Jr.
G. H. McClean
J. A. McCormick
E. A. Miller
W. M. Miner
E. J. Murphy
J. F. Musselman
Howard Newman
F. M. Nortan
David Parkhill
L. M. Peabody
Robt. E. Peck
C. R. Place
W. S. Ransom
T. W. Reynolds
D. R. Richardson
C. M. Ripley
C. H. Seymour
J. F. Siegel
Harry Stokes
R. D. Sutter
H. A. Terrell
H. W. Tinker
G. T. Travis
L. H. Tripp
F. VanWinkle
H. R. Watson
E. K. Webster
H. S. Wheeler
E. V. D. Whitmore
E. S. Williams
C. W. Willcox
C. E. A. Winslow
Paul Wunderlich

LADIES

Mrs. A. S. Armagnac
Mrs. R. P. Bolton
Mrs. W. T. Braemer
Mrs. Bromwell
Mrs. W. H. Carrier
Mrs. A. E. Carpenter
Mrs. J. D. Cassell
Mrs. F. T. Chapman
Mrs. B. C. Davis
Mrs. F. K. Davis
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PAPERS

OF THE

TWENTY-FIRST ANNUAL MEETING

January 19, 20 and 21, 1915

CCCLXII

PRESIDENT'S ADDRESS, 1915

OPERATING COSTS

BY S. R. LEWIS

The responsibility of designing heating and ventilating equipments for buildings in this country is really in the hands of comparatively very few men. These men are overworked, as a rule, being crowded to complete each plant in hand as quickly as possible, and by economic conditions they are forced to take up new plants as soon as the old ones are off the boards.

When the designing engineer has more to do with the operation of the plants he designs, there will be an improvement in design. I speak from experience, and believe that my experience is not unique.

An examination of the Society's proceedings discloses many descriptions of new plants, but little information as to their operation or as to the costs of installation or maintenance.

I venture, therefore, to present an analysis of some data acquired covering three installations of some years' standing.

In 1912 two large factories were designed by the same architects, one in Toledo and one in Detroit. The writer designed the heating equipment for both plants.

The character of the construction is identical. There are no basements but there are some tunnels provided under the first floors for air ducts, service pipes, wiring, etc. The buildings are of reinforced concrete construction with solid concrete floors, mushroom type, and 12 inch brick curtain walls. The glass is set in tight, steel frames extending practically from floor to ceiling, and from column to column. The ratio of glass to exposed wall is approximately three to one. The roof is a concrete slab with a cinder fill and tar above.

The Toledo building is heated and ventilated by an all indirect system, equipped with automatic temperature and humidity control, the humidifying being by means of steam jets. There is no direct radiation whatever, except in a few toilet and service rooms.

The Detroit building is heated entirely by direct radiation, about one-half of the radiation being placed on the side walls and one-half on the ceiling. Great care was taken, however, in placing the radiation on the side walls to provide for a very liberal circulation of air behind it. The Detroit building has no automatic temperature control, although good hand regulation is obtainable by shutting off parts of the radiation.

Both plants are equipped with efficient, two-pipe, vacuum systems.

The Toledo plant is unique in its design to the extent that the blast heating surface is arranged at the bases of the vertical flues and so proportioned that much the same effect is obtained every day as would be obtained by having direct radiators in the various rooms, since gravity indirect heating is always in effect whenever there is any steam in the radiation. The theory in the design was that the Toledo plant should be economical, comparing with direct radiation by reason of this gravity effect, while not open to the objections inherent with direct radiation when placed against the outside walls. These objections are that the direct radiation interferes with the benches of the workmen, causes local overheating, and is not economical of fuel, since there is an opportunity for a large amount of radiant heat to enter directly the outside wall without appreciably affecting the temperature of the room. The air is handled by steam power, and the cost of air handling is included in the fuel cost.

The Detroit plant, with its direct radiation, is, of course, heated whenever supplied with steam.

In order that some idea may be obtained of the relative costs of operating the two plants, a careful record was made of the fuel consumed during the season of 1913-14.

The following is the governing data:

	TOLEDO with ventilation	DETROIT no ventilation
Exposed glass surface	39,520 sq. ft.	13,980 sq. ft.
Exposed wall surface	7,904 sq. ft.	2,796 sq. ft.
Exposed concrete column surface	7,680 sq. ft.	3,600 sq. ft.
Exposed roof surface	45,880 sq. ft.	29,358 sq. ft.
Exposed ground floor surface	45,880 sq. ft.	29,358 sq. ft.
Contents	2,460,500 cu. ft.	704,592 cu. ft.
Floor area	178,800 sq. ft.	56,965 sq. ft.
Blast radiation	12,983 sq. ft.	
Direct radiation	negligible	8,905 sq. ft.
Air delivered per minute	138,000 cu. ft.	
Boiler capacity	500 H. P.	125 H. P.
Cost of coal per season	$3,009.00	$952.00
Fuel cost for heating and ventilating per 1,000 cu. ft. of contents per season.	$1.22	$1.35
Same per thousand sq. ft. of floor space per season	$16.82	$16.73

So far the evidence is favorable to a blast system as indicating that a large, well built, factory building can be heated and ventilated with an efficient, all indirect plant for less cost per thousand cubic feet of space per season and for nearly the same cost per thousand square feet of floor space, per season, as it can be heated alone for by plain direct radiation.

Such a conclusion on the data given is not fair, however, and the reason for the apparent greater economy of the Toledo plant lies in the more economical construction of the building; that is, the Detroit building being but two stories high loses heat through the floor of the first floor and through the ceiling of the second floor, whereas the Toledo building, being four stories high, has two intermediate stories which only lose heat around their edges. This advantage is sufficient in the instance under consideration to make a favorable showing for the blast system.

If, however, we compare the cost per hundred thousand heat units lost per hour (from the theoretical computations using the same factors for each building for zero outside) we find that the Toledo plant cost 65 cents where the Detroit plant cost but 42.8 cents.

It is apparent that owners should consider more earnestly than is apparently their custom, the effect of cold exposure in heating cost.

I believe we can conclude fairly that an increase of 35 per cent. in the fuel cost of heating alone for the addition of good ventilation, is a justifiable expenditure and that the difference in price will prove to be more nearly 35 per cent. than 100 per cent. as is often loosely stated.

The following extracts are made from letters written by the owner of each building in March, 1914, in order to show that both plants were heating the buildings adequately and satisfactorily:

TOLEDO

"Our tenants are very enthusiastic over the heating conditions of the building. The building has been at 70 degrees at 7 o'clock each morning throughout the heating season when the outside temperatures during the day-time have been as low as 3 and 4 degrees below zero.

The building is occupied by various industries, among them the following:

Self-measuring pumps and special tools.
Die castings.
Mittens.
Cloaks and suits.

Milk bottle caps.
Wire ties.
Bath cabinets.
Auto tire pumps.
Rain water shifters.
Spark plugs.
Restaurant.
Eye glass lenses.
Advertising and publishing.
Electrotyping.

The clothing and mitten manufacturers employ a great many women. We have yet to have any complaint from any of the tenants about the building being insufficiently heated. Indeed about a month ago, during the severe cold weather, we made a test of the conditions immediately next to the glass surfaces of our most exposed room—the temperature in the center of the room was 70 degrees; the temperature one foot from the glass was 69 degrees."

DETROIT

"Our heating system, we are very glad to say, has proven quite adequate to take care of us in the severest weather. As to advising whether the two floors equalize in radiation, this we have not followed very closely. There has been no complaint and it would be probably hard for us to determine, as we have a number of firepots on our main floor and the heat from these would probably make some little difference."

The Detroit plant makes automobile radiators and sheet metal accessories.

When buildings are heated by central station steam systems, it is easy to determine the cost of heating them. At Aurora, Ill., the Y. M. C. A. building, heated continuously, of a reasonably good character of construction, gives us the following information:

The plant cost for heating, on the basis of a sliding scale price tor steam, as follows—winter 1912-13:

		Percentage of total each month.
October	$ 43.12	6%
November	77.52	13%
December	84.36	14%
January	140.22	23%
February	100.32	18%
March	90.20	15%
April	57.62	9%
May	14.63	2%
Total	$607.99	100%

The total heat loss per hour is 608,000 B.t.u. computed for an outside temperature of 10 deg. F. $\frac{(608,000)}{966} = 639$ lbs. of water per hour.

If the season were cold throughout, we would theoretically require (24 hours per day) $\times$ (210 heating days) $\times$ (639 lbs. per hour) = 3,220,560 lbs. of steam per season. The plant did condense 1,013,720 lbs. of steam, as of course the maximum condensation occurred but a very small part of the total time.

We may apparently safely deduce from this that instead of figuring arbitrarily 16 hours per day, as is a general custom, instead of 24 hours per day, we can figure the 24 hours and then multiply the result by a percentage. This percentage for the latitude of Chicago is $\frac{1,012,720}{3,220,560} = 31.4$, say 32 per cent.

By using the monthly percentages given above, the cost of heating for any month is obtainable after the total season cost has been computed.

The theoretical loss from the building was based on the following general factors, without any percentages added for points of the compass or leakage.

"K" constant for 1 sq. ft. of various building exposure surfaces, representing the B.t.u. loss per hour per degree difference between the inside and outside temperatures:

BRICK WALLS:

Furred (K).	Bare (K).
8" — .46	8" — .23
12" — .33	12" — .21
16" — .27	16" — .19

Cement floor on ground, K = .31.

Top floor ceiling with shallow attic, K = .15.

Single window, K = 1.20.

Single skylight, K = 1.50.

These are published, more elaborated, by the American Blower Co. of Detroit.

CCCLXIII

THE CENTRIFUGAL FAN*

THE DEVELOPMENT, PERFORMANCE AND SELECTION OF THE STEEL PLATE AND MULTIVANE TYPES

BY FRANK L. BUSEY

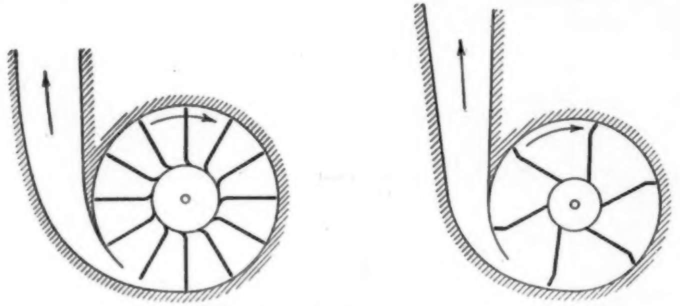
The term "centrifugal fan" is used to designate that form of construction commonly known as our steel plate or multiblade type of blower as distinguished from the positive blower type such as the Root or Connersville blowers. They are used to work against comparatively low pressures—ordinarily of only a few ounces, or at most about 16 ounces. The centrifugal fan is composed of a rotating arbor or plate, to which are attached a series of blades. When this drum is rotated, the columns of air between the blades are given a centrifugal motion, so delivering the air from the center to the circumference of the blades.

The centrifugal fan, in different forms, has been built for over two hundred years, and even as far back as 1847 a Mr. Buckle presented before the Institute of Mechanical Engineers data for the design of fans which does not differ greatly from the practice of the present time. In 1872 the first complete treatise on the theory of the centrifugal fan was published by Daniel Murgue, Engineer to the Colliery Company of Besseges, and this theory is still accepted by our best authorities on fan performance. These earlier fans were probably used for the most part for mine ventilation, and later for the ventilation of tunnels. At first they were used simply for exhausting and were not fitted with a housing. In this way only the centrifugal velocity was used, and no benefit was derived from the rotational velocity.

The earlier fans of which we have any record were developed during the latter half of the last century, a general idea of some

* The subject matter in this paper was prepared by Mr. Busey for a talk which he gave to the members of the Illinois and New York Chapters at their monthly meetings last winter. He afterwards wrote this paper for presentation to the Society and had completed it just previous to his death. It was sent to President Lewis by Mrs. Busey a short time after Mr. Busey had passed away.

of the more common types being given by the accompanying figures. One of the most extensively used of the early fans was the Guibal fan shown in Fig. 1. This fan was invented about sixty years ago for use in mine ventilation, and was constructed in a great variety

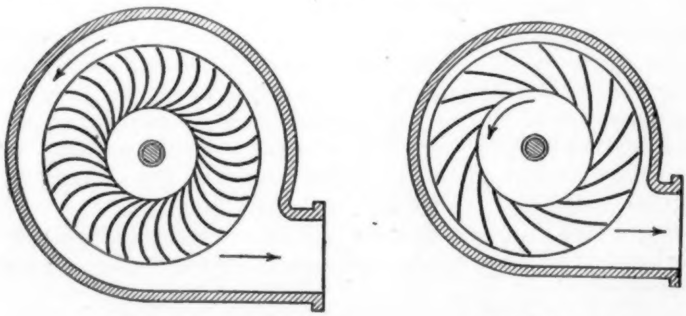


The Guibal Fan and Chimney.

Fig. 1.

of forms by different designers. The one distinctive feature of the Guibal type of fans is the Guibal chimney or discharge-tube on the outlet of the fan. This is a feature still used in fans for mine work, in order to reduce the velocity of discharge and so lessen the loss of head due to discharging at a high velocity.

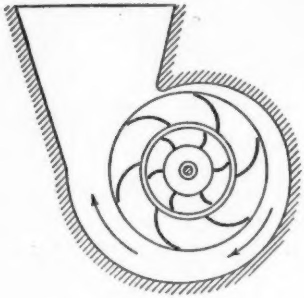
In the majority of cases the Guibal fan was built with the heel of the blade bent forward, in accordance with the theory of Daniel Murgue. This tended to lessen the loss by shock at the heel of the blade. Many of the early fans were built with a circular housing rather than with the spiral or scroll shape used later, although the Guibal fan was built with both styles of housing.



D'Anthonay Fan. Farcot Fan.

Fig. 2.

In Fig. 2 are shown two of the early types in which the blades were curved, the D'Anthonay fan being built with both forward and rearward curved blades. Both of these fans seem to have been built entirely with the circular housing, so losing all benefit from the centrifugal action of the columns of air within the wheel. Only the velocity of discharge was available, and static pressure could be obtained only by the use of a cone-shaped outlet as shown in Fig. 1.



The Capell Fan.

The Rankine Fan.

Fig. 3.

Two fans that have been extensively used in Europe, each embodying special features in its design, are shown in Fig. 3. The Capell fan, invented by Mr. Geo. M. Capell in 1883, is peculiar in that it consists of two rotors, one within the other, having the same axis and revolving at the same angular velocity. Between these two rotors is a circular sheet metal drum, with openings in its surface for the passage of air from the inner to the outer wheel. The total area of these openings was made equal to the area of the fan inlet. The blades all curve backward, or opposite to the direction of rotation. The Rankine fan is built after a design suggested by Professor Rankine in 1857. This design depends on the spiral or scroll shape of the housing to reduce the velocity of discharge rather than depending on the chimney outlet.

The most efficient of the early fans was the Rateau, the published results of tests on different fans of this style showing mechanical efficiencies of 70 to 75 and even approaching 80 per cent. The lines of this fan follow very closely the laws of good fan design, unusual care being taken to reduce as far as possible the internal losses in the fan. Due to its peculiarities of design, this fan would be a more expensive one to build than most of those now on the market The efficiencies quoted are those given in some of the older records, without accompanying description of the methods used in making

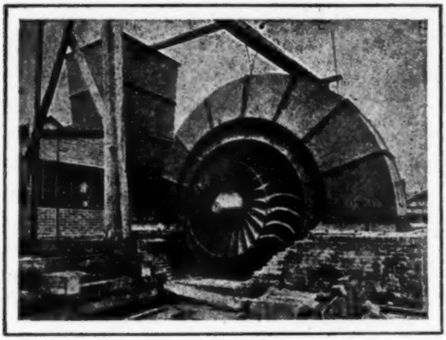
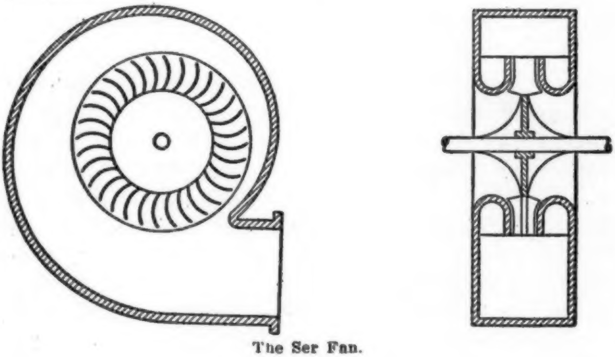


Fig. 4.

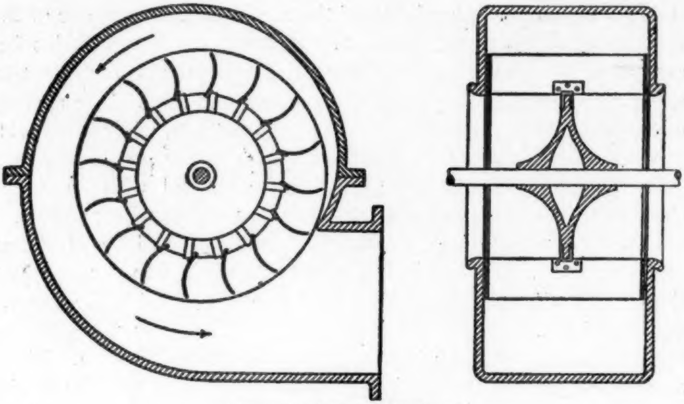
the tests. Fig. 4 shows one of those fans used for exhausting from a mine, where the inlet connection to the mine has not yet been completed. The form of the outlet chimney may be noted from the photograph.



The Ser Fan.

Fig. 5.

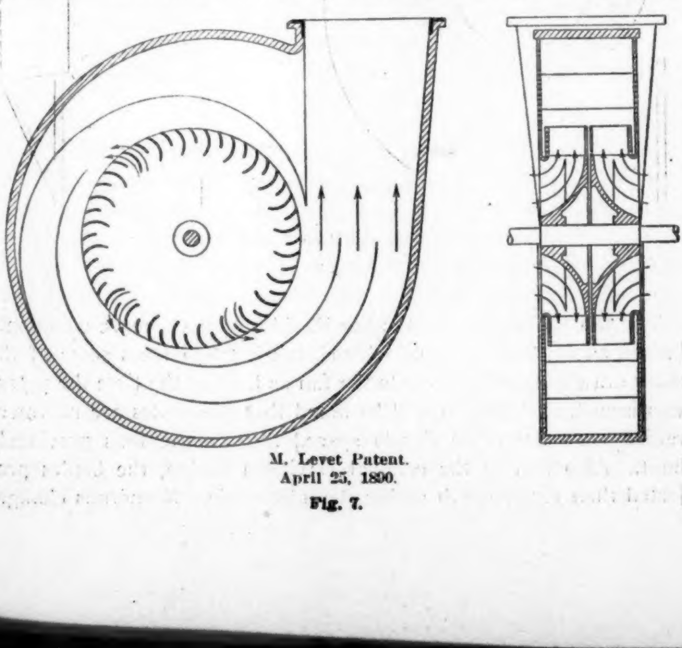
One of the first multivane type fans using short curved blades was that designed by Professor Ser of Paris about 1884. The rotor consists of a central revolving plate, on both sides of which, at the edge, are mounted a series of forward curved vanes. Thirty-two of these vanes are mounted on each side of the plate. There are two special features of this which are worthy of notice. One is the unobstructed inlet chamber, and the other the fact that the tip of the blade is advanced beyond the heel. These are both features given special prominence in the patent granted the Davidson fan, which was invented some fifteen years later.



Fournier & Cornu Patent.
February 18, 1896.
Fig. 6.

What has the appearance of having really been the direct ancestor of our modern narrow blade multivane type of fan is shown by the two views of the fan patented by Fournier and Cornu in 1896. This design, as well as the one shown in Fig. 7, are both French patents which are not on file at the United States Patent Office, and have never been published except by title by the French Patent Office.

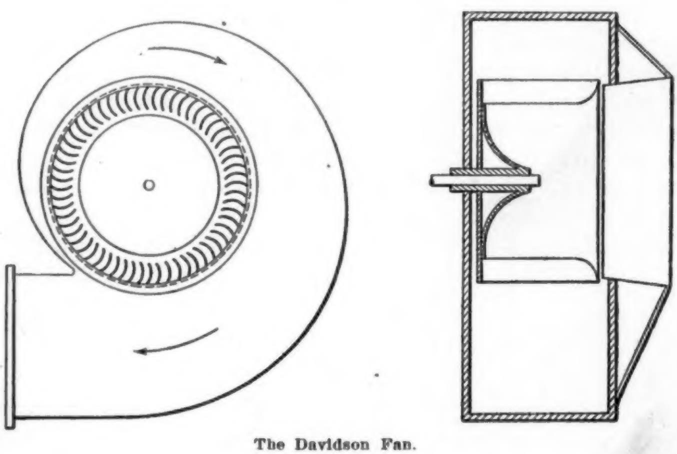
The fan shown in Fig. 6 consists of a wheel having curved blades mounted on a central revolving disc. It will be noted that the unob-



M. Levet Patent.
April 25, 1890.
Fig. 7.

structed inlet, one of the claims of the Davidson patent, is one of the features of this fan. According to the patent, the blades of this fan are twice as long as their radial dimension while the Davidson patent specifies blades as being at least three times as long. The spacing of the blades is here equal to the depth of the blade, while the Davidson fan uses about 2/3 of the depth. That is, the Davidson patent calls for longer blades more closely spaced, and with an intake chamber relatively greater in diameter than the fan here shown.

Fig. 7 shows sectional views of the fan patented by M. Levet in April, 1890. This is in some respects similar to the one just shown although no claim is made in the patent covering the proportions used. One noticeable feature about this fan is the generous use of guide vanes, both in the inlet and outlet of the fan. These constitute one of the principal claims of the patent. The blades as here shown have the tips advanced beyond the heel of the blade, but the patent claims the right to curve them in either direction and in any manner required to meet special cases.

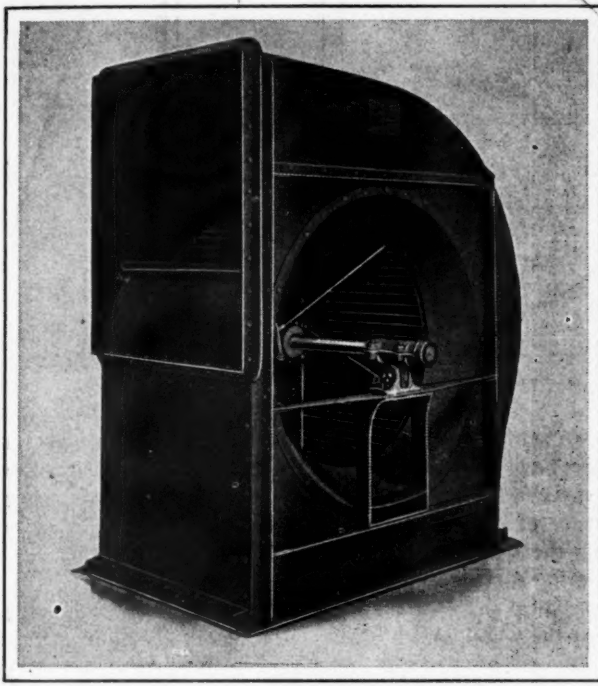


The Davidson Fan.

Fig. 8.

The fan shown in Fig. 8 is the Davidson patent, more commonly known as the Sirocco fan. This sketch is taken from some of the older drawings and represents the fan as built at the time the patent was issued, in 1900. It will be noted that the blades are narrower radially and are more closely spaced than in the fans previously built. Also that in the original Davidson design, the blades projected their entire depth within the inlet circle. Numerous changes

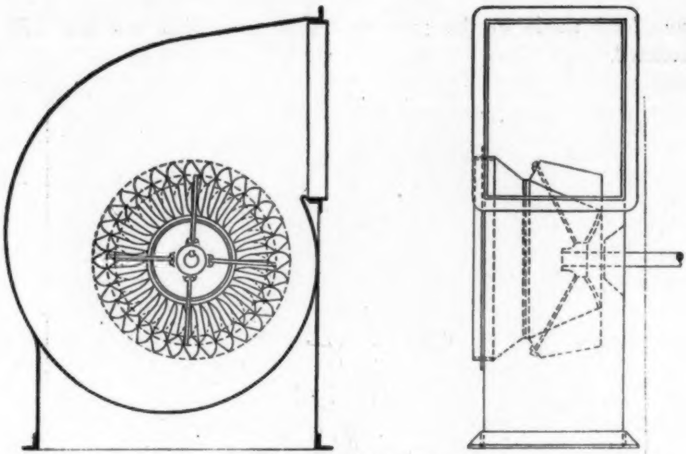
have been made in the general design since this fan was first invented.



Single Inlet American "Sirocco" Fan—Full Housed.
Fig. 9.

In the modified design as employed in this country the wheel used is in most respects similar to the first one just shown, but as will be seen from Fig. 9, the blades do not project within the inlet circle. It will also be noticed that the pipe connection at the fan outlet comes to about the center of the fan housing, so that a much larger pipe with the same size outlet is used than shown on the original design. That is, the pipe connection is increased to practically fifty per cent. larger than the discharge area by extending the outlet connection below the point of cut-off as here shown.

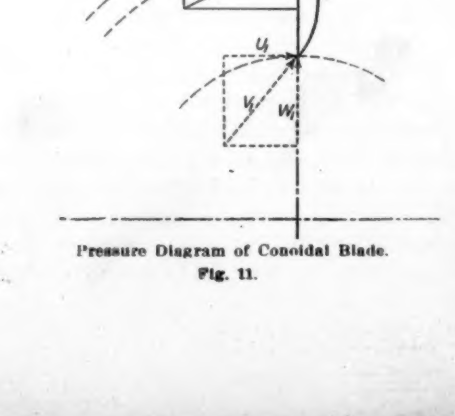
The Niagara Conoidal fan as built at Buffalo is the most recent development in the design of multivane curved blade fans. As will be noted from Fig. 10, this fan embodies two special features; first, the diverging cone effect is brought back into the fan housing,



Niagara Conoidal Fan.
Fig. 10.

giving an actual lower outlet velocity and consequent higher static pressures; and secondly, the wheel is entirely different in its design from anything previously built. The name was derived from the prevalence of conical sections used in the design, the blast wheel forming the frustrum of a cone and the blades being curved over the tapering surface of a cone.

This conical shape of the blades is clearly shown in the sectional detail of the wheel, which gives a view of the blades from the inlet



Pressure Diagram of Conoidal Blade.
Fig. 11.

side and looking toward the back plate. This shows clearly the greater depth of the blades at the back side of the wheel, and also that the wheel is of greater diameter at the front than at the back side. These wheels may be made either single or double, the double wheel being simply two single wheels mounted back to back.

As will be noticed from Fig. 11, each blade is a portion of a conical surface, being generated on a cone by a generatrix not parallel to the axis. This is the only design in which proper regard has been paid to the angle of incidence at the heel of the blade, and the only fan in which this angle is varied across the width of the blade to care for the difference in radial velocities. The securing of the proper angle of incidence reduces the loss by shock at the heel of the blade.

It will be noticed from the parallelogram that the radial velocity is greater at the back of the blade than at the front, while due to the smaller diameter at this point the tangential velocity is less. The resulting velocities, as shown by the diagonals in each case, make different angles with the vertical, and the heel of the blade is made to conform to this angle. The effect of this is greatly to reduce the loss by shock of the air entering the blades and thus to increase the pressure and reduce the power consumption. As will be later explained the centrifugal force varies with the depth of the blade, increasing as the difference of the squares of the radii. By making the diameter and consequently the tangential velocity less at the back of the wheel than at the front, the difference in centrifugal velocity is so balanced as to give an even pressure across the entire tip edge of the blade, which results in a maximum efficiency.

THE RELATION BETWEEN STATIC, VELOCITY AND TOTAL PRESSURE, AND BETWEEN STATIC AND TOTAL EFFICIENCY OF CENTRIFUGAL FANS

Frequent reference has been made to *static pressure* and *velocity pressure,* and also the conversion from velocity to static pressure at the fan outlet. In fan work the air is delivered against a certain definite resistance of the system, due for instance to ducts and heaters, and this is termed static resistance. It is then necessary for the fan to develop a definite static pressure sufficient to overcome this resistance of the system. In addition to the *static* pressure, the air has imparted to it a certain *velocity* pressure due to its velocity on leaving the fan outlet, this velocity being dependent on the amount of air handled and on the area of the outlet.

The total energy imparted to the air is composed of the static pressure of the system and the energy of discharge corresponding

to the velocity pressure, or velocity head as it is termed in hydraulics. *The total pressure is the sum of the velocity pressure at the fan outlet and the static pressure of the system,* and is the pressure upon which the performance and efficiency of the fan is usually based. In the case of an exhaust or draw through system, the static head on the fan should be taken as the difference in static pressure at the inlet and outlet of the fan, one being positive and the other negative.

The relation between static and total pressure and between static and total efficiency in fan work is not generally understood. Many engineers as well as fan builders consider only the total efficiency of a fan, even when the resistances are estimated in static pressures. The static pressure is the potential energy developed by the fan, and is the energy available for performing useful work in overcoming the frictional resistance of the system. The velocity pressure is the kinetic energy used in moving the air and is a measure of the air quantity handled, but until it is converted to static pressure is not available for overcoming resistance.

Thus it is seen that the greater the static pressure developed at the fan outlet in proportion to the velocity pressure, the more effective will be the performance of the fan in overcoming frictional resistance, and therefore the greater the *static efficiency.* Since the velocity and therefore the velocity pressure varies inversely as the area of the fan outlet, and since the sum of the static and velocity pressures are constant (being equal to the total pressure), it is evident that the larger the outlet of a fan the greater will be the static pressure developed.

The efficiency of a fan is the ratio of the power output to the power input, or the ratio of the work done in delivering a certain amount of air against some known pressure to the work done by the engine or motor used in driving the fan. We may have two different efficiencies, either *static efficiency* or *total efficiency,* depending on whether we are considering static or total pressure. Since the total pressure is greater than the static pressure, the total efficiency is greater than the static efficiency, and for this reason the efficiency generally referred to in fan performance is based on the total pressure.

It has been seen that a high static pressure and a correspondingly high static efficiency are of vital importance in fan performance. A fan having a high total but a low static efficiency may in most cases be less desirable than another having a high static efficiency, even though the total efficiency be less than that of the first fan. Since in the majority of installations the principal work performed by the fan is in overcoming the static resistance of the system, a fan

giving a high static efficiency means a low power consumption, and at least where power is being bought it means a saving in dollars and cents.

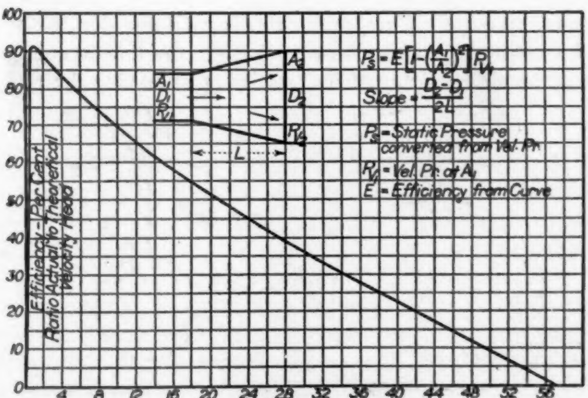
Although but few fan builders are accustomed to mention the static efficiency of their product, the static efficiency may be readily calculated when the ratio of static to total pressure at rated capacity of any special fan is known. Thus if the static pressure is 80 per cent. of the rated total pressure, the static efficiency will be 80 per cent. of the total efficiency. Or expressed as formulæ, the efficiencies are:

$$\text{Static eff.} = \frac{0.0001565 \times Q \times \text{static press. in inches}}{\text{H.P.}}$$

$$\text{Total eff.} = \frac{0.0001565 \times Q \times \text{total press. in inches}}{\text{H.P.}}$$

where Q equals the quantity of air in cubic feet per minute, the pressure is expressed in inches of water, and H.P. represents the horse power input.

Thus it is seen that the ratio of static to velocity pressure at the fan outlet is of the greatest importance in fan selection. This ratio varies as the square of the fan capacity, at constant speed, and bears a definite experimental relationship to the efficiency of the fan. As already mentioned, where the velocity and consequent velocity pressure at the fan outlet is very high, a part of this velocity pressure may be converted to static pressure and the desired ratio between



Efficiency of Diverging Nozzles.

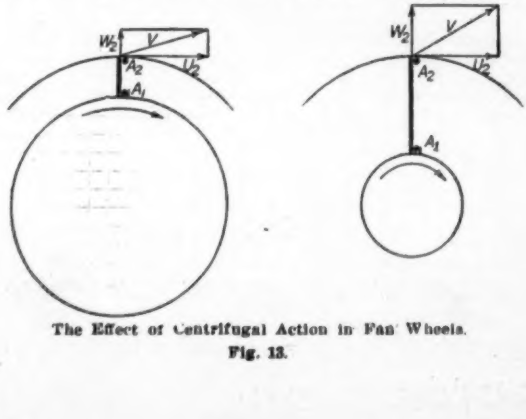
Fig. 12.

the two obtained, by using a diverging cone or chimney outlet on the discharge of the fan. The efficiency of this conversion depends on the slope of the sides of the cone, and is never the full one hundred per cent. That is, there is always some loss of head in converting from velocity to static pressure.

As will be noted from a study of Fig. 12, a diverging cone or nozzle to give the maximum conversion from velocity to static pressure with the least possible loss must be constructed with an easy slope and not with an abrupt flare. The diagram here shown indicates what may be termed the efficiency of diverging nozzles having different slopes to the sides. We note that with a 10 per cent. slope, or a slope to each side of say one inch in ten, the efficiency of conversion is 70 per cent. That is, while there will be a certain drop in velocity pressure due to the enlargement in cross-section of the pipe, only 70 per cent. of this drop will be converted to static pressure in the cone and the balance will be lost. While as high as 90 per cent. conversion may be obtained, the cone would have to be longer than is generally found practical and a cone with 10 per cent. slope gives good average results. If made with a length of twice the smaller diameter, and the area of the large end twice that of the small end, a diverging nozzle will have a slope of approximately 10 per cent.

RELATIVE PERFORMANCE OF THE STRAIGHT AND CURVED BLADES AND OF THE STEEL PLATE AND MULTIVANE TYPES OF FANS

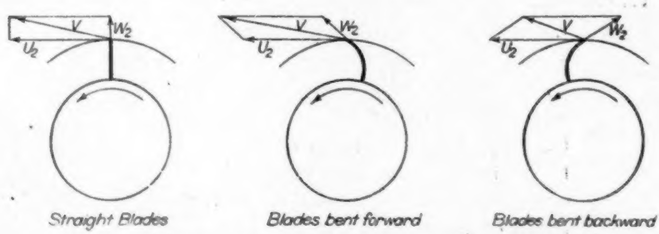
In any centrifugal fan there are two separate and independent sources of pressure. First, pure centrifugal force due to the rotation of an enclosed column of air. Second, the kinetic energy contained in the air by virtue of its velocity upon leaving the periphery



The Effect of Centrifugal Action in Fan Wheels.
Fig. 13.

of the fan rotor. The amount of centrifugal force imparted to the air depends largely upon the ratio of the tangential or rotational velocity of the air leaving the periphery of the rotor to the tangential or rotational velocity of the air entering the fan at the heel of the blade.

The effect of the centrifugal force of the column of air between the blades of a fan is frequently overlooked, since it is difficult to think of the air as having weight. The action of the particles of air within the fan wheel may be illustrated by the two diagrams in Fig. 13. If we consider a particle of air located at A_1, at the heel of the blade, when the wheel is rotated this particle will be thrown to the tip of the blade, and the centrifugal force will give it a certain velocity W_2. But at the same time, the particle has imparted to it a certain rotational or tangential velocity U_2. The resulting velocity V represents the force actually imparted to the particle of air as it leaves the tip of the blade. A comparison of the two figures will show why a fan with longer blades—if they are both straight—will give a greater pressure than will a short blade fan at the same speed. That is, the pressure for a given peripheral speed is increased by increasing the relative length of the blades.



Relative Effect of Different Shaped Blades.

Fig. 14.

The method used for overcoming this fault of the short straight blade is to make it curved. The effect of so doing is shown by the diagram Fig. 14, the effect on the velocity by curving the blades forward or backward being indicated by the parallelograms. It will be noticed that when the blade is bent forward, a pressure greater than that corresponding to the peripheral velocity is obtained. This results in the same pressure with a lower speed than would be necessary with the straight blade when using the same wheel diameter. On the other hand it is sometimes desired to direct connect a fan to a high speed unit without developing the corresponding high pressure. This is accomplished by bending the

blades backward, so obtaining a pressure less than that corresponding to the peripheral velocity.

Centrifugal fans may be roughly divided into two classes, fans having rotors with straight radial blades, and fans having rotors with blades curved with reference to their direction of rotation. Curved blade fans have quite diverse characteristics depending on whether they are curved forward or backward with reference to their direction of rotation. The mathematical theory of the radial blade fan is very completely and clearly discussed in Prof. Carpenter's book on Heating and Ventilation. The amount of total pressure developed by a straight blade fan may be mathematically determined, but in the case of a curved blade fan this can only be known by means of actual tests.

The amount of pressure developed by a straight blade fan may be determined by means of the formulæ.

$$P_s = \left(\frac{U_2}{V_0}\right)^2 \left[e + (1 - R^2) - f\left(\frac{V_1}{U_2}\right)^2\right] - \left(\frac{V_2}{V_0}\right)^2$$

$$P_v = \left(\frac{V_2}{V_0}\right)^2$$

Where P_s = static pressure at fan outlet.
P_v = velocity pressure at fan outlet.
V_0 = velocity corresponding to unit pressure.
V_1 = velocity of air entering fan inlet.
V_2 = velocity of air at fan outlet.
U_2 = peripheral velocity of wheel.
R = ratio inlet to wheel diam.
e and f are coefficients varying with the ratio of height to width of fan outlet.

When the flow of air through the rotor of a fan is partially obstructed the centrifugal effect in the rotor produces a compression corresponding to the centrifugal force, which is known as static pressure. On the other hand, the kinetic energy of the air leaving the periphery of the rotor must first be converted largely into potential energy in the form of static pressure before being serviceable. This conversion from kinetic energy or velocity into potential energy or static pressure is ordinarily accomplished in the scroll formation of the fan housing. A still further conversion is often secured, where the velocity leaving the outlet is high, by means of a diverging nozzle on the outlet of the fan.

The velocity of the air leaving the tip of the blades and the corresponding velocity pressure is greatly in excess of that ordinarily

required in the piping system, and at the same time the static pressure is too low. By enclosing the wheel in a casing having a properly designed scroll, this velocity is reduced and a part of the velocity pressure is converted to static pressure. Since the static pressure due to the wheel will vary as the difference of the squares of the rotational velocities at the periphery and inlet, it is evident that the shorter the blade the greater must be the dependence on the scroll shaped housing to obtain the desired static pressure. For this reason the proper design of the housing is of greater importance in the case of a short blade multivane type of fan than with the older styles.

The standard steel plate fan is essentially a straight blade fan, as compared with the later styles of the short curved blade multivane type, although, as already shown, when the tips of the blades are bent either forward or backward the fan will have different characteristics from one with strictly straight blades. This fan as ordinarily built does not give as high an efficiency as the multivane type, owing to the fact that it is designed for large capacity rather than for high efficiency. But if these long blade fans are built according to special design they may be made to give greater efficiency than can be obtained from the curved short blade fans. This calls for a tall narrow fan with the inlet diameter smaller than that used on the standard fan. It may be readily shown that there is a certain diameter of inlet that will give maximum economy of operation. If the diameter is increased the loss by impact at the heels of the blades is increased as the square of the diameter, and the loss by entrance is decreased as the fourth power of the diameter. The opposite holds true in case the inlet diameter is decreased.

COMPARATIVE EFFECT OF BLAST WHEEL PROPORTIONS UPON THE EFFECT OF STRAIGHT BLADE FANS OPERATING AT THE SAME CAPACITY AND PRESSURE.

Ratio of Dia. Inlet to Dia. Wheel at Perip.	Per Cent. Relative Diameter.		Per Cent. Relative Width.	Per Cent. Relative H.P.	Per Cent. Relative Speed.
	Wheel.	Inlet.			
0.700	82.0	91.9	108.9	112.3	123.0
0.650	93.2	97.5	102.5	104.0	109.5
0.625	100.0	100.0	100.0	100.0	100.0
0.600	106.9	102.6	97.5	96.7	92.7
0.550	123.5	108.8	92.1	91.0	78.2
0.500	144.9	116.6	85.9	86.8	64.5
0.450	170.8	123.0	81.3	83.4	53.3
0.400	206.5	132.4	75.5	80.0	43.1
0.350	255.0	142.8	70.1	77.5	34.6

TABLE I

The proper size of the fan inlet depends on the cubic feet of air per revolution handled by the fan. It has been determined both

mathematically and experimentally that the most efficient diameter of inlet is given by the equation

$$D_1 = C \sqrt[3]{\frac{Q}{N}}$$

Where D_1 = inlet diameter in feet.
Q = cu. ft. air per minute.
N = revolutions per minute.
C = is a factor determined experimentally and is practically a constant for all ratios of inlet diameter to wheel diameter.

It may be noted from Table I that the essential factor in the design of straight blade fans is the diameter of the inlet, and that the smaller the inlet as compared to the diameter of the wheel, the greater will be the efficiency attained, but at a sacrifice in capacity. This table is based on the assumption that a value of 62.5 per cent. for the above ratio be used to represent the average standard fan, and the other figures show the comparative values for other ratios. It will be seen from the second and fourth columns that the height of the fan increases rapidly while the width decreases. This means, then, that these special high efficiency fans are tall and narrow,

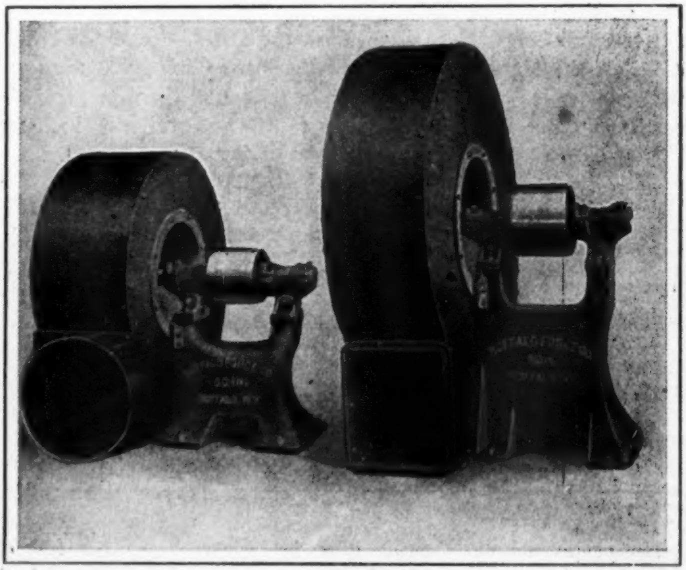


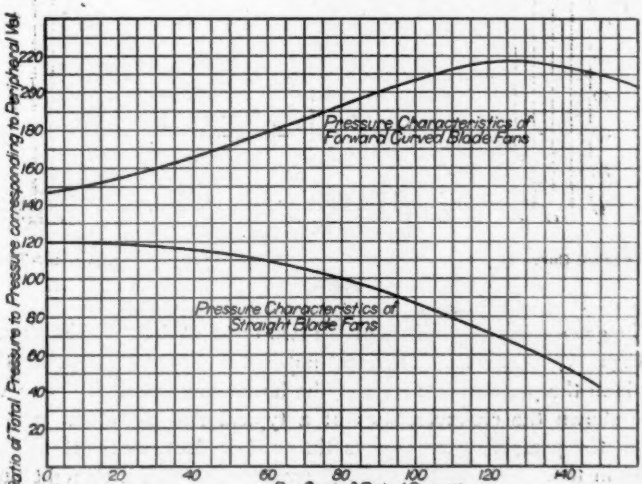
Fig. 15.

which naturally makes them more expensive than the ordinary commercial steel plate fan.

The two fans shown in Fig. 15 illustrate the principles just discussed, one fan being designed along standard commercial lines, while the tall fan is especially designed to give a high efficiency and operate at a low speed. Both of these fans are referred to as fifty-inch fans, because they have the same capacity. This taller fan will make possible a saving of approximately fifteen per cent. in the power bill over the standard fan.

These special tall narrow fans are frequently used for induced draft work, partly because the narrow wheel makes a shorter overhang on the fan bearing, and partly because they may be operated at lower speed and are therefore more suitable for direct connection to steam engines.

It will be noted from the diagram Fig. 16, that the pressure characteristics of the fan with forward curved blades are quite different from those of a fan having straight blades. Although this peculiarity of the two types is often overlooked and perhaps not generally understood, it is nevertheless of the greatest importance and

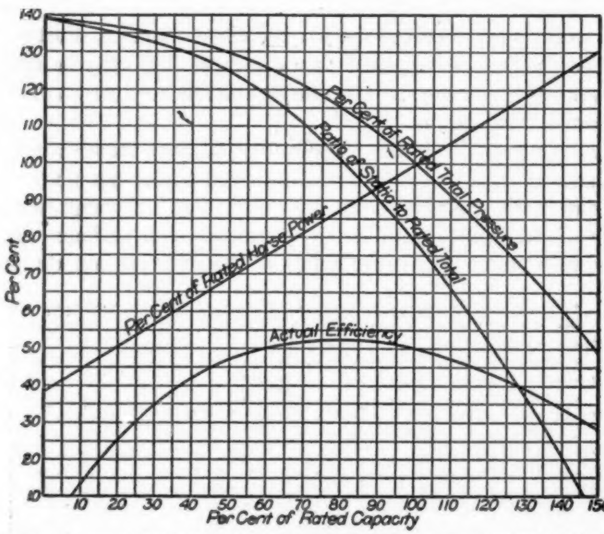


Pressure Characteristics at Constant Speed.

Fig. 16.

should be considered in the selection of a fan. We see that with a straight blade fan the pressure tends to build up as the load on the fan is reduced, while operating at constant speed.

Thus, if such a fan be used to supply forced draft to a boiler, and due to the thickening of the fuel bed the discharge from the fan should be throttled, the pressure will increase. This is just exactly the condition that would be desired. On the other hand, with a forward curved blade, as the air delivery is decreased the pressure would also fall off. Another case where this peculiarity is of importance is when a fan is required to operate part of the time at a considerable underload, yet a definite pressure must be maintained. For such a case the straight blade fan should be used.

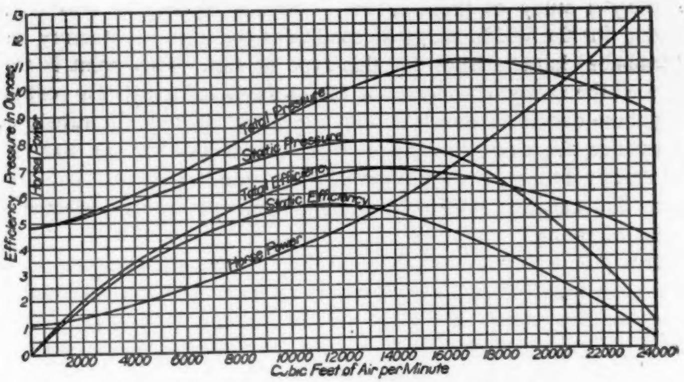


Performance Curve of Buffalo Planoidal Steel Plate Blowers. Straight Blade Type.

Fig. 17.

The general characteristics of the two types of fans will be shown by Figs. 17, 18 and 19. The most noticeable difference between the two is in the pressure curves and the horsepower curves. Here we have shown the characteristics of a typical straight blade fan, in which the pressure increases as the load is decreased. It will be noted in this case that the H.P. characteristics give a straight line, this being a peculiarity of the straight blade fan.

Fig. 18 shows the actual test results obtained from a fan with forward curved blades. In making a shop or laboratory test the most accurate results are obtained by attaching to the fan outlet a straight pipe about thirty diameters in length, with means provided at the discharge end for attaching plates having different sizes

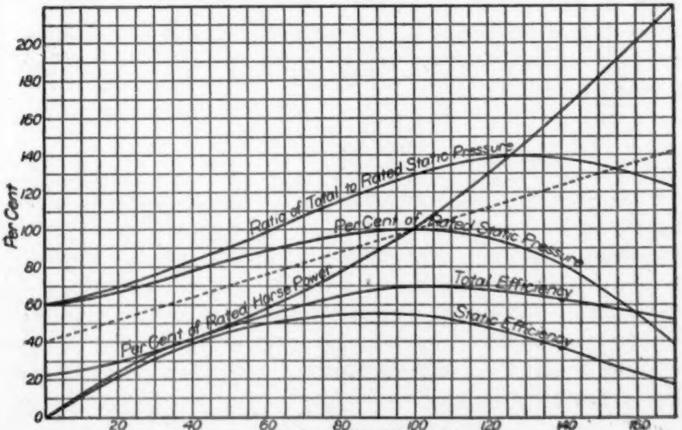


Test of a No. 8 Niagara Conoidal Fan, 450 r.p.m.

Fig. 18.

of openings. In this way, the quantity of air handled may be varied from nothing to the full capacity of the fan, and by this means the performance of the fan at different capacities may be determined. The fan is then operated at some constant speed, and by means of a pitot tube in the discharge pipe about twenty diameters from the fan outlet the pressures developed with each outlet are noted. At the same time power readings are taken in order to compute the horsepower and efficiency.

These values are then plotted and curves drawn as in Fig. 18,



Performance Curve of Buffalo Niagara Conoidal Fans. Forward Curved Blade Type.

Fig. 19.

and by a study of these curves a certain capacity is selected as giving for the widest range of conditions the most satisfactory performance characteristics for this type of fan. This point is then termed the rating of this size of fan at the speed used. By redrawing these curves as shown in Fig. 19, in per cent. of the rated performance, the characteristics of this style of fan for any size may be readily found for any other than the rated capacity. The ordinary fan capacity tables are evolved from the values selected as point of rating.

It will be seen from this diagram, Fig. 19, that the pressure curves for a fan with forward curved blades are entirely different from the ones already shown in Fig. 17 for the straight blade type. At no load the total pressure is 47 per cent. of the rated, while with the straight blade fan the ratio is 140 per cent. The dotted line is the H.P. curve reproduced from the diagram of a straight blade fan. The most important point to be noted here is the great difference between these two lines beyond the point of 100 per cent. capacity. We see how rapidly the horsepower goes up with an overload, with the total pressure also increasing. This explains why any fan with short forward curved blades is so liable to overload the motor, if by any chance the static resistance of the system should happen to be less than estimated. For instance, if a fan should draw its supply of air through the by-pass during the summer, instead of through the heater, as ordinarily designed, the resistance of the system may be decreased sufficiently to increase the quantity of air handled as much as 15 per cent. The increase in power consumption for a straight blade fan would be about 8 per cent., but with a forward curved blade such as here considered the increase in power would be about 19 per cent.

FAN SELECTION

It has been the custom until recently for the fan builders to issue capacity tables of their fans based on total pressure and giving the fan performance at its rated point—that is, the point of highest total efficiency. The growing custom of specifying static resistance has led to the publication of static pressure capacity tables for many of the fans now on the market, and in some cases these tables are arranged to give the fan performance at other than its rated point. Such tables have been issued for the Sirocco and for the Niagara Conoidal fans. As these tables may appear somewhat puzzling to one not familiar with their use, a brief explanation of their application in the selection of a fan may prove of interest.

CAPACITIES OF BUFFALO NIAGARA CONOIDAL FANS (TYPE N) UNDER AVERAGE WORKING CONDITIONS—AT 70° F. AND 29.92 INCHES BAROM.

Fan No.	Mean Dia. of Blast Wheel.	Area of Outlet. Sq. Ft.	⅜ In. Total Press. or 0.217 oz.			½ In. Total Press. or 0.288 oz.			⅝ In. Total Press. or 0.360 oz.			¾ In. Total Press. or 0.433 oz.		
			R.P.M.	Vol.	H.P.	R.P.M.	Vol.	H.P.	R.P.M.	Vol.	H.P.	R.P.M.	Vol.	H.P.
3	15⅝	1.31	413	1490	0.13	478	1720	0.19	533	1930	0.27	585	2110	0.35
3½	18⅛	1.79	354	2030	0.17	409	2350	0.26	457	2620	0.37	501	2870	0.48
4	20½	2.33	310	2650	0.22	358	3070	0.34	400	3430	0.48	439	3750	0.63
4½	23½	2.95	276	3360	0.28	318	3880	0.43	356	4340	0.60	390	4750	0.80
5	26⅛	3.64	248	4150	0.35	287	4790	0.53	320	5350	0.74	351	5870	0.98
5½	28¾	4.41	225	5020	0.42	260	5800	0.65	291	6470	0.90	319	7100	1.19
6	31⅜	5.25	207	5970	0.50	239	6900	0.77	267	7710	1.07	292	8450	1.41
7	36½	7.14	177	8130	0.68	205	9400	1.05	229	10490	1.46	251	11500	1.92
8	42	9.33	155	10610	0.89	179	12260	1.37	200	13700	1.91	219	15020	2.51
9	47	11.81	138	13450	1.12	159	15520	1.73	178	17340	2.41	195	19000	3.18
10	52	14.58	124	16580	1.39	143	19160	2.14	160	21400	2.98	175	23460	3.93
11	58	17.64	113	20070	1.68	130	23180	2.58	146	25900	3.60	160	28390	4.75
12	63	21.00	104	23880	2.00	119	27590	3.08	133	30820	4.29	146	33780	5.65
13	68	24.65	95	28040	2.35	110	32370	3.61	123	36180	5.03	135	39650	6.63
14	73	28.68	89	32520	2.72	102	37550	4.19	114	41950	5.84	125	45990	7.69
15	78	32.80	83	37330	3.13	96	43100	4.80	107	48160	6.70	117	52790	8.83
16	84	37.32	78	42470	3.56	90	49040	5.47	100	54790	7.62	110	60060	10.1
17	89	42.14	73	47950	4.01	84	55370	6.17	94	61800	8.60	103	67800	11.4
18	94	47.24	69	53750	4.49	80	62060	6.92	89	69340	9.64	98	76010	12.7
19	99	52.63	65	59890	5.00	75	69160	7.71	84	77260	10.8	92	84700	14.2
20	105	58.32	62	66360	5.56	72	76640	8.54	80	85600	11.9	88	93850	15.7

Static pressure is 77½% of total press.

TABLE II

The sample capacity tables here shown are taken from the tables of the Niagara Conoidal fan, and are based on the performance curves shown in Figs. 18 and 19. Table II is a reproduction of what might be termed the rated capacity table, giving the capacity, speed and horse-power of the different sizes for different total pressures, when operating at what has been selected as the rated point on the curves. This is approximately the point of highest efficiency. These tables are similar to practically all of the fan tables heretofore published and require no particular explanation.

Table III is one of a set which, like the recently published tables for the American Sirocco, may at first appear puzzling to one inexperienced in their use. While Table II gave the performance of all sizes at the most efficient point only, this one gives the performance for one size only, but all along the curve on both sides of the most efficient point.

There is one point in each pressure column at which the fan will give its best efficiency, indicated on the table by the bold-faced figures. For each pressure there is a certain velocity through the outlet that will give the best efficiency, as for instance, 2,500 velocity for one inch static, and this holds true for all sizes. It may often be a matter of expediency to operate a fan at other than this most efficient point, and we will consider this question more in detail later.

A question frequently asked concerning these tables is how will one find the performance of a fan at any other than the pressures given in the tables. When using the regular tables of rated capacities we are accustomed to work directly across the table for each size, and find that for different pressures the speed and capacity vary as the square root of the pressure and the horsepower, as the three halves power. In the case of these new tables, we cannot work directly across, since each line represents a constant capacity. By working diagonally across the table, that is, by finding in the capacity column a new air quantity proportioned as the square root of the pressures and then following across the table to the desired pressure column, we will find the required speed and horsepower.

As an example, we will assume it is desired to handle 31,000 cubic feet of air per minute at 1¼ inch static and we wish to know the speed and horsepower of a No. 10 fan for this purpose. The table gives the data required at one inch, so we multiply 31,000 by the square root of the ratio of 1 inch to 1¼ inch and find the corresponding capacity when reduced to a one-inch basis to be 27,700 A.P.M. From the table, the speed at one inch will be 232 R.P.M. and require 7.91 H.P. The speed for the required capacity

NO. 10 NIAGARA CONOIDAL FAN (TYPE N).
Capacities and Static Pressures at 70° F. and 29.92 In. Barom.

Outlet Velocity Ft.-Min.	Capacity Cu. Ft. Air Per Min.	Add For Total Press.	½ In. S.P.		¾ In. S.P.		1 In. S.P.		1½ In. S.P.		2 In. S.P.	
			R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.
1400	20410	0.122	**164**	**2.92**	206	4.61	243	6.59	308	11.1		
1500	21870	0.141	163	3.13	204	4.78	240	6.83	305	11.5		
1600	23330	0.160	164	3.42	202	5.02	238	7.05	302	11.8	357	17.0
1700	24790	0.180	165	3.74	**201**	**5.30**	235	7.28	299	12.1	353	17.5
1800	26240	0.202	166	4.13	**200**	**5.61**	233	7.50	295	12.4	350	17.9
1900	27700	0.225	168	4.55	200	6.01	232	7.91	293	12.7	**347**	**18.3**
2000	29160	0.250	171	5.04	200	6.48	**231**	**8.32**	291	13.0	343	18.7
2100	30620	0.275	174	5.56	201	7.00	231	8.77	288	13.5	340	19.2
2200	32080	0.302	177	6.12	203	7.54	230	9.31	286	13.9	338	19.6
2300	33540	0.330	180	6.76	205	8.16	231	9.92	285	14.4	336	20.1
2400	34990	0.360	183	7.43	207	8.86	232	10.6	**284**	**15.0**	332	20.6
2600	37910	0.422	190	8.95	213	10.4	235	12.1	282	16.3	329	21.8
2800	40830	0.489	198	10.7	219	12.2	240	13.9	283	18.1	**327**	**23.3**
3000	43740	0.560	206	12.7	226	14.3	246	16.0	285	20.1	326	25.0
3200	46660	0.638	215	14.8	234	16.7	251	18.3	288	22.4	327	27.4

Note: Bold-face figures indicate point of highest static efficiency.

TABLE III

at 1¼ inch pressure will vary from the above directly as the capacity, and the power as the cube of the capacity. The No. 10 fan will then handle 31,000 A.P.M. against 1¼ inch static at 260 R.P.M. and require 11.1 H.P.

SPEED AND HORSE-POWER OF NIAGARA CONOIDAL FANS, AT VARIOUS CAPACITIES AND ONE INCH STATIC PRESS.

Outlet Velocity Ft.-Min.	Add Total for Press. Inches	NO. 6.			NO. 7.			NO. 8.		
		Capacity, Cu. Ft.-Min.	R.P.M	H.P.	Capacity, Cu. Ft.-Min.	R.P.M	H.P.	Capacity, Cu. Ft.-Min.	R.P.M	H.P.
1600	0.160	8400	397	2.54	11430	340	3.46	14930	298	4.51
1700	0.180	8920	392	2.62	12150	336	3.57	15860	294	4.66
1800	0.202	9450	389	2.73	12860	333	3.72	16800	291	4.86
1900	0.225	9970	387	2.85	13570	332	3.88	17730	290	5.06
2000	0.250	10500	385	3.00	14290	330	4.08	18000	289	5.33
2100	0.275	11030	385	3.16	15000	330	4.30	19600	289	5.61
2200	0.302	11550	384	3.35	15720	329	4.56	20530	288	5.96
2300	0.330	12070	385	3.57	16430	330	4.86	21460	289	6.35
2400	0.360	12600	387	3.82	17150	332	5.19	22400	290	6.78
2500	0.390	13120	389	4.10	17860	333	5.59	23330	291	7.30
2600	0.422	13650	392	4.36	18580	336	5.93	24260	294	7.74
2800	0.489	14700	400	5.00	20000	343	6.81	26130	300	8.90
3000	0.560	15750	410	5.76	21430	352	7.84	28000	308	10.2
3200	0.638	16790	419	6.59	22860	359	8.97	29860	314	11.7
3400	0.721	17850	432	7.00	24290	370	10.3	31720	324	13.5

NOTE: Highest static efficiency for each fan is at 2,000 ft. per min. outlet velocity.

TABLE IV

Perhaps the most important function of this new set of tables is to enable one to select different sizes of fans for any desired duty, and to tell at a glance the corresponding speed and horsepower. The table here shown is a composite, giving the performance of three sizes of fans, each operating against a static resistance of one inch. As mentioned before, when operating at one inch static the highest static efficiency will be attained with an outlet velocity of 2,000 feet per minute.

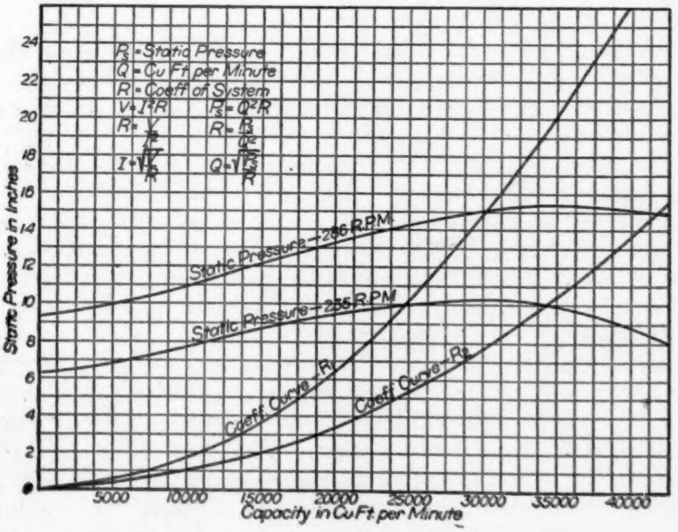
These outlet velocities tend to aid in the selection of a fan, inasmuch as it is good practice to use certain velocities with installations of different character. Thus in the case of schools, where quiet operation is absolutely essential, an outlet velocity of from 1,800 to 2,200 should be used. As the static resistance of such an installation will seldom run over one inch, we note that this is also the point of highest efficiency. In the case of a shop where some noise is not so objectionable, where high duct velocities will be used, and where the static resistance will probably approach two inches, an outlet velocity of perhaps anywhere between 2,800 and 3,400 may be used. For special cases we may use either higher or lower velocities, according to the requirements to be met. But for school-house work one would seldom be warranted in using over 2,200 outlet velocity.

By a comparison of the three sets of data in Table IV, it will be seen how we may select either one of the three fans for the

same duty. Assuming that we are required to supply 15,000 cubic feet of air per minute at one inch static, the bold face figures show that we may use the No. 6 at 400 R.P.M.; the No. 7 at 330 R.P.M.; or the No. 8 at 298 R.P.M. The No. 7 will come the nearest to the most efficient point, requiring less power than either of the others.

In case we expect to use high duct velocities and are interested in first cost rather than economy of operation, the No. 6 would be the fan to use. If low operating cost is the most essential factor to be considered and moderate velocities are allowable, the choice should fall on the No. 7. But in case very low velocities are absolutely necessary, even at a higher initial cost and a slight increase in operating cost, the No. 8 fan should be used.

One of the questions frequently asked by any one using these tables for the first time is how is it that we expect to get two different capacities at the same speed, when we have always been taught that the speed varies as the capacity. As will be seen from the table, each size of fan will deliver two different quantities of air at the same speed with a constant static resistance. But we must also note that the total pressure is different for each case. This means we would be operating at two different points on the



Relative Performance of a No. 10 Niagara Conoidal Fan.

Fig. 20.

performance curve. We have already noticed from Fig. 19 that one of the peculiar features of the pressure curve of this fan is what we might call the hump, or high spot. Now this curve was drawn for a varying capacity but a constant speed, so that over a certain range there will be two capacities for the same speed and pressure, hence the two capacities in the table.

Perhaps the simplest way to illustrate this point would be to show that each installation has a certain definite coefficient of resistance and that the quantity of air handled is a direct function of this coefficient and the static resistance of the system. As shown in Fig. 20, the relations holding in the case of a fan are directly comparable to those with which we are familiar in the case of electricity. We have, first, the relation of voltage drop or voltage resistance across the terminals is equal to I^2, a measure of quantity, times R, the coefficient of resistance, this product being known as the I^2R loss. In the same way, we have the static resistance on the fan which is equal to the square of the air quantity times the coefficient of resistance of the system. By simply changing the position of the factors in this equation we have the coefficient of resistance of any system is equal to the static pressure divided by the square of the air quantity. Also the air handled will be a definite fixed quantity, equal to the square root of the static pressure divided by the coefficient of the system. The static pressure is a constant for any certain installation, being a measure of the static resistance against which the fan is to operate. Thus the static resistance on a fan drawing through six sections of Buffalo Standard Pipe Heater would be 0.57 in.

On the diagram, Fig. 20 are drawn two curves representing the coefficient of resistance of two different installations. The horizontal curves represent the static pressure curves for two different speeds. The points of intersection of each diagonal with the horizontal curves indicate what will be the performance of the fan at either speed for either of the two installations. That is, with a coefficient of R_1 and a speed of 235 R.P.M. the fan will deliver 24,800 cubic feet of air per minute against one inch static, and it will not deliver any other quantity. If a different installation is considered with coefficient R_2, this same fan will deliver 34,200 cubic feet of air per minute against one inch static, with the same speed of 235 R.P.M. That is, each fan will deliver, within certain limits, two air quantities at the same speed and static pressure, but not on the same installation. If we increase the air quantity we increase the velocity and consequently the friction and static resistance. We have changed the conditions of our system and will have

a different coefficient R, with the result that we will operate at a different point on the performance curve.

We note that with a constant coefficient, as for instance R_1, if it is desired to handle more air it is necessary to operate at a higher pressure. Thus, if we speed up to 286 R.P.M., we will develop a static pressure of 1.5 inches and deliver 30,100 cubic feet of air per minute. The ratio of this new static pressure to the square of the air quantity is still the same as that for the first case. That is, the value of R_1 is the same.

CONCLUSIONS

The practice of specifying the static resistance against which a fan is to operate and then selecting a fan that will give a high static pressure at the fan outlet and a high static efficiency is to be commended. Where only the total efficiency of a fan is given, but the ratio of the static to the total pressure is known, the static efficiency may be calculated by multiplying the total efficiency by the ratio of the pressures.

As has been mentioned before, in making the selection of a fan the choice lies between the older type of straight blade fan, and the newer multivane type. The impression has of late been prevalent among engineers that the multiblade type is inherently more efficient than the straight blade fan, but we have seen that such is not necessarily the case. A properly designed radial vane fan may be made to give even higher efficiencies than can be obtained with a multiblade type, but as already shown such a fan will be much larger in diameter and narrower in width.

The real advantage of the multiblade type is the attainment of fairly high efficiency in a more limited space, which makes it of great commercial value for certain classes of work. In case an increasing pressure is desired with an increasing resistance, it has been shown from the pressure curves that the forward curved type is not applicable unless operated beyond its most economical point. On the other hand, it is frequently desired to maintain a constant or increasing pressure with an increase in capacity. In such cases the forward curved type is the only fan capable of accomplishing the desired results. Another important advantage of the multiblade fan is the fact that its higher speed makes it more suitable for direct connection to motors, or at least gives better pulley ratios than may be obtained with radial blade fans.

CCCLXIV

ENGINE CONDENSATION

With Particular Reference to Heating with the Exhaust

BY PERRY WEST, MEMBER

After remarking at one of our recent New York Chapter meetings, that the subject of cylinder condensation was somewhat shopworn and divested of the proper interest as a subject for some proposed tests, I was asked by our Secretary to prepare this paper upon the subject. Failing to prevail against his desire in this connection, you will appreciate the position which I am in.

I remember, while going into the subject myself some time ago, however, that several very interesting points came up which may be worthy of notice. I remember also of having to brush a great deal of dust off of my elementary thermodynamics and that the subject looked quite different and seemed to hold a fancy, strange to my last recollections of such matters.

If you will pardon my doing so, therefore, I shall start this talk about some of these little things upon which we frequently build so much and then sometimes forget that they are the underlying foundation of our structures.

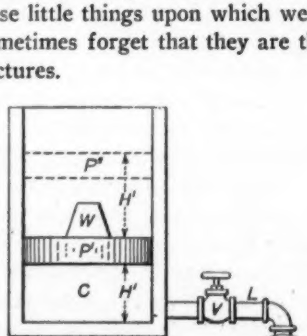


Fig. 1.

Figure 1, shows a cylinder (C) closed at one end and open at the other.

(P) is a tightly fitting frictionless piston free to move up and down and (W) is a superimposed weight bearing on the piston. (L) is a pipe line leading from a steam supply (B) under pressure. If now, the valve (V) be opened, steam will enter and raise the piston and weight, and when closed again, the piston will stop at a point (P′), such that the total weight just balances the total pressure on the underside of the piston. Now assuming that the weight (W) and the piston are of exactly the same weight, if we remove the weight (W) the piston will rise to point (P″), so that the original volume of the steam (as contained in the cylinder when the piston was at (P′), will be practically doubled and its pressure per unit of area will be practically halved. During this process while the piston is passing from (P′) to (P″) no additional steam is admitted to the cylinder, but an amount of work is done measured by the product of the weight of the piston and the height (H′).

This, as we all know, is what is ordinarily known as the work of expansion of the steam. Since there is no heat being supplied to the cylinder during this period the heat equivalent of this work must be supplied by the steam confined within the cylinder itself.

The only way in which heat can be extracted from steam is either by reducing the temperature or condensing some of the steam. If the steam happens to be superheated above its normal temperature at the original pressure, the first operation consists in the steam giving up enough of this extra heat to equal the amount of work done or down to the point of saturation of the steam.

If the heat equivalent of the work done is in excess of this superheat, the steam will then begin to cool and then to condense and continue to do so until the sum of the superheat and the heat of condensation equals the heat equivalent of the work, that there will be no condensation from this source.

As stated above the processes which we have just been discussing are those attending what is ordinarily known as the expansion period of an engine's cycle of operations. It may be of interest, before proceeding further, to return to a more minute consideration of the first operation of bringing the steam into the cylinder. Returning to Fig. 1, and assuming that the volume of the steam space in (B) is exactly equal to the volume in the base of the cylinder when the piston is at (P′), and that there is no steam being generated for the moment; it is easy to see that the process of filling the cylinder is one of the expansion of the steam just the same as in the case of the expansion from (P′) to (P″). The work performed and the heat equivalent of the resulting condensation will be about twice as much, but otherwise the actions will be analogous.

We are not accustomed to considering this admission period as a process of expansion, but upon close analysis, it is found to be, with more or less modification, no less so than the regular expansion period of the cycle.

If the steam should be generated in the boiler at exactly the same rate at which it is used in the cylinder, the process would, of course, be one of displacement all along the line, the same as if the fluid were water. The fact, however, that steam is generated in practice, at a regular rate and used in pulsations, means that the rate of use during this period is very much higher than the rate of generation. Take an engine cutting off at 1/4 stroke and having a connecting rod of eight crank lengths, the steam is only being admitted for about one-third of the time.

Taking into consideration, too, the fact that the steam cannot get into the cylinder any faster than the piston moves out of its way, and that the rate of motion of the piston is zero at the beginning of admission and gradually increases towards cut-off, it is readily seen that the rate of admission at cut-off is many times the average during admission and many more times the average for the entire cycle. Neglecting the angularity of the connecting rod, the rate of motion of the piston is equal to the rate of the crank pin, multiplied by the cosine of (90°) minus the angle which the crank stands from the dead center.

For the above example the rate at cut-off would be .866 times the rate of the crank, while the average rate up to the point of cut-off is only about one-half the rate of the crank travel. The rate of admission at cut-off would, therefore, be 1.73 times the average rate during the admission and 5.19 times the average rate for the entire cycle.

At this rate, about five units of steam would be used while one unit is being generated so that four out of these five units used would be forced into the cylinder by expansion of the steam in the boiler and piping. Of course this does not apply to such a marked degree, except throughout the later stages of admission. As a matter of fact, during the very first stages of admission steam is generated faster than it is used, but this condition soon changes and rapidly increased in the opposite direction. It is very evident, therefore, that there is a great deal of expansion and expansive work which takes place outside of the engine cylinder itself.

This expansion work outside of the cylinder does not cause any appreciable net condensation, however.

This may be best understood by considering all of the processes through which the steam passes on its way from the boiler to the

cylinder. In the first place, just before admission the pressure throughout the system outside of the cylinder, is at its maximum.

As admission takes place the pressure drops and reaches its minimum just before cut-off.

In this interval expansion takes place and some condensation usually results. After cut-off the steam outside of the cylinder is compressed by the new steam being generated, and is finally brought up to its original pressure just before admission again.

This process of compression adds heat to the steam and re-evaporates whatever condensation might have occurred during its expansion. The heat equivalent of the work of admission, which is taken out of the steam in the cylinder, is put in at the other end by the displacement process and cannot, therefore, be again charged to another account.

To return now to the work of expansion in the cylinder, I wish to speak of one more peculiar phase of these phenomena, which appealed to me as a most refreshing little spark, lighting the way through the passages of this somewhat dreary subject.

In our first consideration of the expansion in the cylinder we disregarded the time element altogether. In other words, we simply said that a certain weight lifted through a certain distance by the steam, meant a certain amount of work of expansion done by the steam, and that this work of expansion by the steam is exactly equal to the external work done. Now the question naturally arises as to what effect the time element has upon this process. If the expansion be allowed to take place very slowly it would appear that there could be little question as to the truth of the above assumption.

As we increase the speed of the operation, however, we begin to inquire as to what force is employed in moving the steam itself. We naturally assume a condition, therefore, where the piston is first held in place and then suddenly moved, by an external force, at just such a speed as to keep ahead of the steam at its fastest rate of travel, without allowing the steam to exert any pressure whatever on the piston. We then ask the question as to what has become of the work of expansion of the steam as it has exerted none on the piston, which is the only moving part absorbing work. In other words, does steam when expanding from one pressure to another lower pressure always perform work? And as a consequence, does it always lose heat and undergo a corresponding partial condensation?

The answer to these questions as far as the above example is concerned, is that as long as the piston is moving the work of expansion of the steam is put into the kinetic energy necessary to produce the

high velocity of its particles, and that a corresponding quantity of heat is given up, and a corresponding amount of condensation takes place.

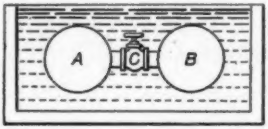


Fig. II.

Scientists tell us that if two closed vessels such as (A) and (B) in Figure 2, having a connection with a stop cock (C) be immersed in water and gas be allowed to expand from (A) (under pressure) into (B) which is empty, that upon equalization of pressures and temperatures, the temperature of the whole will not be changed due to this expansion. Here we have an instance of expansion of gas without any external manifestation of the work done.

The explanation is that the gas expanding from (A) does work on its particles to put them in motion and consequently loses heat. These same particles in being brought to rest against the sides of vessel (B) and against the stationary particles in this vessel generate an equal amount of heat which produces the final equilibrium of heat for the apparatus as a whole. Here, after the gas is at rest, we have no external work remaining, and consequently no heat change, but still at the same time the gas has expanded to twice its volume.

The explanation seems to be that if there is any net work done, either externally or on the gas itself, there will be a corresponding amount of heat loss; otherwise there will be no heat loss. This feature, of the expansion of steam without heat loss, will be spoken of later in connection with the condition of steam after passing a pressure reducing valve.

With the above ideas refreshed, we will endeavor now to analyze the different processes surrounding the passage of steam through an engine.

On starting with a cold cylinder, the feed (consisting of steam and water) strikes the cold surface of the ports, cylinder walls and piston, and as a consequence a considerable amount of heat is transferred from the feed to these colder bodies, with a corresponding cooling of the water and a partial condensation of the steam. This process continues up to the point of cut-off, but of course the percentage of condensation is heaviest at the beginning, owing to the iron being at its lowest temperature and the amount of surface in contact being greatest per unit of feed admitted.

The same process continues after cut-off, but as the steam is cooled due to its expansion and drop in pressure, the temperature difference between steam and iron is not so great and the rate of condensation from this source is reduced. A new factor of condensation is introduced, however, during this period, due to the work of expansion.

As expansion continues, the pressure and temperature of the steam drop, and as this ensues the temperature of the steam soon drops below that of the cylinder walls, which have just been exposed to the admission feed, but, of course, is still above that of the cylinder walls which are being newly exposed as the piston moves forward. This produces a complicated set of conditions whereby heat is transferred from the steam to the cooler surfaces and from the hotter surfaces to the steam. These two opposing factors finally balance at some point of the expansion, after which the greater transfer of heat is to the steam causing a net re-evaporation of some of the condensate.

At release, and from thence throughout exhaust, the transfer of heat is all to the steam causing a re-evaporation of most of the condensate, not produced by the work of expansion and by radiation.

This is not nearly so true when an engine is first started, as it is after the same has been running long enough for the cylinder to assume its normal running temperature. The first impacts of heat, which re-occur with each stroke, are utilized in altering the temperature of the iron immediately in contact with and adjacent to the steam. These waves immediately begin to penetrate the walls, however, and soon raise the temperatures of the outside surfaces in contact with the air. As soon as the temperatures of these outside surfaces are above that of the surrounding air we have a transfer of heat by radiation, etc., from same to the air. If the iron of the cylinder was susceptible to large instantaneous heat changes we would have a condition where these outside surfaces would alternately be at the temperature of the admission feed and then at the temperature of the exhaust with a variable range of temperatures between.

The natural resistance of the iron to such sudden temperature changes causes these heat impacts to travel out slowly and to produce a greater part of the temperature changes, throughout any particular stroke, on a very thin skin of the iron. As the engine continues to run from a cold start, however, the heat continues to penetrate the iron until the outer surfaces assume a more or less uniform temperature; approximating an average between the temperatures of the admission and exhaust steam.

If the cylinder is lagged with a non-conducting material the surface of this will assume a more or less uniform temperature, which

will, of course, be below that of the iron, depending in amount upon the efficiency of the insulation.

When the condition of uniform temperatures is attained there will be a continuous loss of heat to the surrounding bodies, which must of course be supplied by the steam and usually results in condensation.

Further than this there is very little heat loss from the steam outside of that corresponding to the amount of work done; as it can readily be seen that if the heat extracted from the steam during admission and the earlier stages of expansion was not all returned during exhaust, or otherwise accounted for by work or radiation, the whole system would continue to rise in temperature indefinitely, which is not the case.

Starting now with 100 per cent. of steam at the boiler we will endeavor to determine the resulting quality at exhaust.

A certain percentage of entrained moisture is usually allowed for, to be deducted from the steam passing to the piping system.

From this must be deducted the percentage of condensation due to radiation from the piping system. This may be allowed for on the basis of 15 per cent. of the heat radiated from bare pipes, for well insulated conditions, and proportioned for the quantity of steam handled per hour, per square foot of radiating surface.

A further deduction must then be made for the condensation due to the work.

And lastly, a deduction for the condensation due to radiation and other direct heat losses from the cylinder and adjacent parts of the engine.

All other exchanges of heat may be neglected since as we have seen they do not remain as factors in the determination of the final condition of the steam at exhaust.

In the practical solution of this problem for any particular case, the following data and suggestions may be of use:

The percentage of entrained water in the steam leaving the boiler may be easily measured, or under favorable conditions, it may be assumed at from 2 to 3 per cent.

The condensation in the piping due to radiation may be estimated from the temperature differences between steam and air, the square feet of radiating surface per pound of steam handled per hour and the nature of the insulation. Multiply the square feet of surface per pound of steam handled per hour by the temperature difference, and this product by 30, and divide by the latent heat of steam at boiler pressure and the result will be the percentage of condensation for well insulated pipes.

SIZE AND TYPE OF ENGINE	I.H.P.	LBS. STEAM PER I.H.P. HOUR	BOILER PRES. LBS. GA.	TOTAL No. EXPANSIONS	PISTON SPEED IN FT.	% MOISTURE IN STEAM	% WATER IN CYLINDER H.P. C.O.	H.P. REL.	L.P. C.O.	L.P. REL.	% STEAM TO JACKETS	ABSORBED BY IRON DURING ADMISSION	GIVEN UP BY IRON DURING EXPANSION	GIVEN UP BY IRON DURING EXHAUST	GIVEN UP BY IRON DURING EXH. L.P. CYL.	GIVEN UP BY IRON DURING EX. L.P. CYL.	RADIATION H.P. CYL.	RADIATION L.P. CYL.	SUPPLIED BY JACKETS H.P. CYL.	SUPPLIED BY JACKETS L.P. CYL.
WOOLF BEAM	267.86	17.73	84.8	9.64	371	2.5	37.6	24.3		17.5	7.1	26.2	12.1	18.3	9.2	11.5	0.5	0.7	2.2	3.3
24½"×3'-5"	267.94	17.39	88.14	9.56	376	2.5	37.7	21.5		14.3	8.3	25.4	14.5	16.1	9.4	9.6	0.5	0.7	2.5	3.7
38"×5'-6"	267.91	19.24	88.95	7.77	379	2.5	34.0	28.0		35.2	None	25.2	6.4	20.2	3.7	22.4	0.5	0.7	0	0
TANDEM COMPOUND	120.30	21.65	86.05	11.48	683	4.0	42.6	25.4		26.3	"	35.6	19.3	20.0	1.8	17.2	0.4	0.6	0	0
15¾"&28½"×51"	149.76	20.88	84.77	11.64	692	4.0	43.6	25.5		26.3	"	36.1	20.1	19.3	2.1	16.4	0.4	0.7	0	0

Interchange of heat between steam & walls of cylinder in % of total heat in feed.

FROM TESTS BY MR. GEO. MAIR 1890 IN
"PEABODY'S THERMO-DYNAMICS OF THE STEAM ENGINE"

PERRY WEST 1914.

TABLE, (I.)

Neglecting the work of compression the percentage of condensation due to the work may be determined by multiplying the absolute indicated horse-power of the engine by 33,000 and then by 60 (to reduce same to its equivalent in foot pounds per hour), and then dividing this result by 778 (to reduce same to B.t.u.), and then subtracting the heat equivalent of the work of admission (which equals the cubic feet of steam handled per hour multiplied by the average absolute admission pressure, and then by 144 divided by 788), and then subtracting the B.t.u. gotten by multiplying the total cylinder feed per hour by the difference in total heat of steam at admission and exhaust pressures, and dividing the remainder by the latent heat of steam at exhaust pressure, then dividing by the total pounds of cylinder feed per hour, and multiplying by 100 to reduce the result to a percentage basis.

The condensation due to radiation, etc., may be estimated theoretically, but might be better gotten from reports of engine tests.

I would refer in this connection, to table No. 1, taken from Peabody's Thermodynamics showing radiation losses as gotten from tests by Mr. Geo. Mair in 1881 and 1882; also to a discussion of the subject by Mr. R. J. S. Pigott, in the July and August, 1910, issues of Engineering Magazine, where a loss of two per cent. for high pressure, and one per cent. for the low pressure cylinders respec-

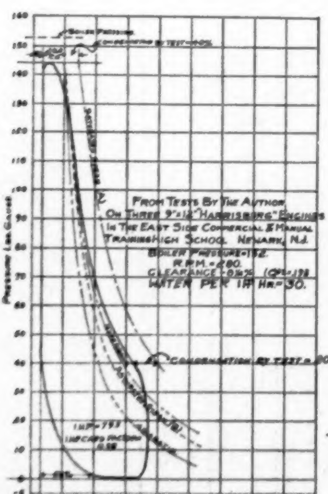


Fig. III.

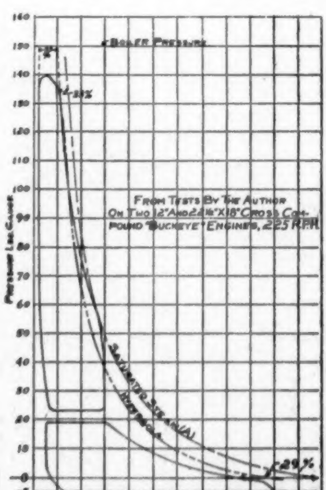


Fig. IV.

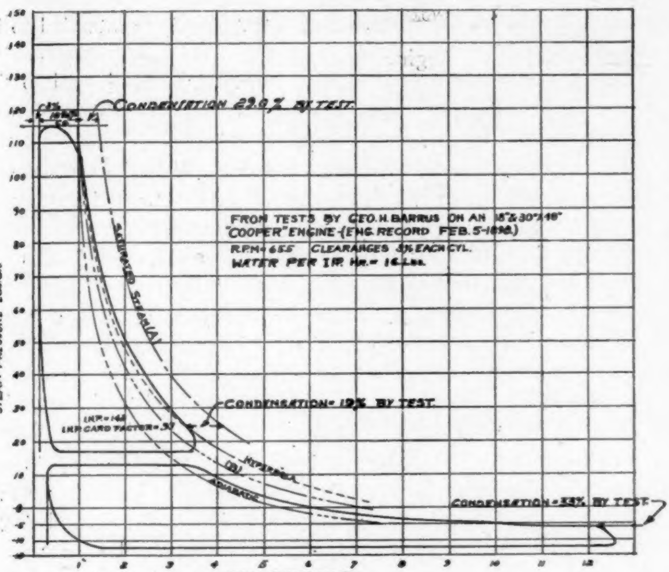


Fig. V.

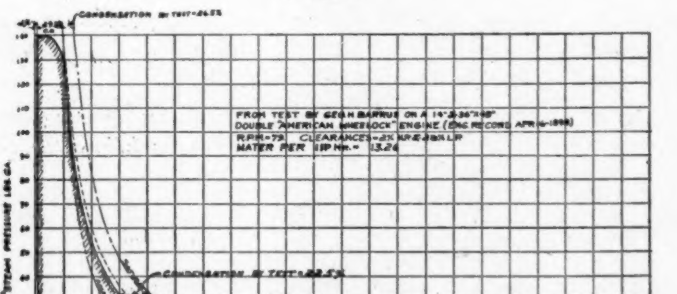


Fig. VI.

tively of the 59th Street New York Subway Station engineers were decided upon.

I would refer to Figures Nos. 3, 4, 5 and 6 as displaying graphically the condition of the feed at various points throughout its passage through the cylinder. These sheets have been made up from

the data on some representative engine tests, and are intended to be self-explanatory.

The saturated curves marked (A) indicate the total feed to the cylinder. The saturated curves marked (B) indicate that portion of the feed accounted for by the indicator at cut-off.

It will be noted from these curves that the higher the economy of the engine and the greater the amount of work extracted per pound of steam, the greater the percentage of condensation at release. This is generally true since the work is usually the largest factor in the production of this condensation.

The percentage of condensation shown by these curves should not be misunderstood to indicate the final percentage of condensation in the mixture which leaves the engine cylinder. As has already been explained, a great deal of the heat transferred to the cylinder, etc., during the early part of the stroke is returned during exhaust.

Unfortunately these curves, and engine tests generally, do not show what happens to the feed during exhaust. The curves serve, however, to show very plainly, in a graphical way, the several stages through which the feed passes between admission and release.

Understanding now all of the process through which steam passes on its way from the boiler to the exhaust outlet of the engine we are prepared to solve any particular problem involving the condition of this steam and to compare its condition with that of low pressure, or reduced pressure, steam for heating purposes.

For use in this connection the various factors representing condensation may be put into the form of formula, as follows:

For condensation in steam pipes:

$$C_1 = \frac{S_1 \times (T_1 - T_2) \times 2}{W_1 \times H_1} \times K_1 \times 100$$

Where C_1 = per cent. of condensation due to radiation from piping.

S_1 = sq. ft. of surface in piping.

T_1 = temperature of steam.

T_2 = temperature of surrounding air.

2 = B.t.u. transmitted per hour per sq. ft. of bare pipe per degree difference.

W_1 = pounds of feed passing per hour.

H_1 = latent heat of the steam.

K_1 = a constant depending upon the efficiency of insulation.

100 = factor to reduce to percentage basis.

For well insulated piping (K) may be taken as 0.15 which reduces the formula to:

$$C_1 = \frac{30\,S_1\,(T_1 - T_2)}{W_1 \times H_1}$$

For condensation due to work in the cylinder:

$$C_2 = \frac{(HP \times 33000 \times 60) - Q_1 + 144 \times P_1 - W_1\,(H_3 - H_4)}{778\,W \times H_2} \times 100$$

Where C = percentage of condensation due to work in the cylinder.

HP = absolute indicated horse-power.

33000 = ft. pounds per minute.

60 = minutes per hour.

Q_1 = cu. ft. of steam handled per hour at admission pressure (as indicated between clearance and cut off).

144 = sq. inches per sq. ft.

P_1 = average absolute admission pressure.

778 = ft. pounds per B.t.u.

H_2 = latent heat of steam at exhaust pressure.

H_3 = total heat of steam at admission pressure.

H_4 = total heat of steam at exhaust pressure.

W_1 = total pounds of cylinder feed per hour.

This formula reduces to

$$C_2 = \frac{2545\,HP - .188\,Q_1 \times P_1 - W_1\,(H_3 - H_4)}{W_1 \times H_2} + 100$$

For condensation due to radiation, etc., from cylinder:

$$C_3 = \frac{D_1 \times L_1}{W_1} \times K_2 \times (T_3 - T_4)$$

Where C_3 = percentage of condensation due to radiation, etc., from the cylinder.

D_1 = diameter of cylinder in inches.

L_1 = length of stroke in inches.

W_1 = weight of feed handled per hour.

K_2 = A constant depending upon the insulation and other variable conditions.

T_3 = average between temperature of admission and exhaust steam.

T_4 = Temperature of surrounding air.

If the condensation in piping is drained away before reaching the cylinder, a further deduction should be made for condensation in the exhaust as follows:

$$C_4 = \frac{W_2 \times (T_5 - T_6)}{W_1 \times H_2} \times 100$$

Where C_4 = condensation in exhaust due to loss of heat in drips.
W_2 = pounds of drips drained away.
T_5 = temperature of drips.
T_6 = temperature of exhaust steam.
W_1 = pounds of feed handled.
H_2 = latent heat of steam at exhaust pressure.

It will be seen from the foregoing that in passing high pressure steam from a boiler through a system of piping and thence through a reciprocating engine, a considerable heat loss is encountered which usually results in a considerable percentage of condensation in the exhaust. I should say that with simple engines this would run between 15 per cent. and 20 per cent. This means, of course, that there is never as much steam available for the heating system as is started with at the boiler, by just this amount of condensation. Besides this the steam is in a very moist condition, due to the presence of this water.

Now on the other hand, if we start with high pressure steam at the boiler and reduce the pressure through a pressure reducing valve, we expand the steam without performing any net work and hence the steam is superheated. The degree of superheat can be estimated by taking the difference between the total heat of steam at the two pressures and figuring how much this will raise the temperature of steam at the low pressure.

In the reduced pressure steam we have, therefore, 100 per cent. of the steam started with at a higher temperature than the steam leaving an engine would be.

This means also that we have more total heat available for use in the heating system per pound of steam generated.

If this heat can be as readily utilized, therefore, these points are in favor of reduced pressure, as against exhaust steam for heating, all other factors being disregarded.

There is much diversity of opinion, however, based on the fact that the moist exhaust steam transmits its heat more readily than the drier reduced pressure steam. The fact remains though, that fewer pounds of the hotter steam will be required at the boiler due both to its greater total heat and absence of condensation losses.

As to low pressure steam, this, of course, should act practically the same as exhaust steam, except that less would be required to be generated and the heat per pound to generate is also less. The efficiency of a low pressure boiler should also be higher, but is frequently less than a high pressure one.

CCCLXV

THE HEATING VALUE OF EXHAUST STEAM

BY DAVID MOFFAT MYERS, MEMBER

In my work with Factory Power Plants the efficient utilization of exhaust steam very commonly comprises a determining factor in the over-all economy of the plant. Consequently the heating value of exhaust steam has received my careful consideration both as to its actual application in practice and also with reference to the calculation of its amount from given data.

To the latter aspect of the problem I will principally direct my remarks. In presenting my first paper before this Society I have endeavored to develop my subject in such form as to make the contents of direct value to contractors and heating engineers for the laying out of systems which shall fulfill the predicted expectations.

It should be needless to state that the weight of steam and moisture issuing from the exhaust of an engine is equal to the weight of the steam and moisture entering the throttle of the engine. We are therefore only interested in *the thermal and physical condition of the fluid mixture as it leaves the engine.* Thus if we could conceive the impossible, i. e., that its condition is the same at the exhaust as at the throttle, then the heating value of exhaust steam would precisely equal that of live steam.

There are only two ways in which heat is abstracted from the steam in passing through an engine and all heat not so deducted appears in the exhaust in the form of steam or hot water. These two ways are:—

1—By conversion of a part of the heat into mechanical work.

2—By dissipation of a small amount through cylinder radiation.

Hence if we know what these are we may quickly compute the heat which remains in the exhaust. We may further compute accurately how much moisture the exhaust steam mixture will con-

tain. Item 2, the radiation, comprises so small a percentage of the total heat in the steam used by the engine that its quantity is negligible. For instance, a 100 horse-power non-condensing engine at full rating may use about $100 \times 30 \times 1189^* = 3{,}567{,}000$ B. t. u. per hour. By allowing a radiating surface of say 25† square feet (a too generous allowance to be on the safe side) and assuming continuous boiler pressure inside the cylinder, the outside air being at 70 degrees Fahr., we may treat the engine cylinder as a high pressure radiator and compute the B. t. u. radiated as follows:—

$3^* \times 25 \times (339† - 70) = 20{,}175$ B. t. u. per hour. Comparing this amount of radiated heat to the total B. t. u. supplied in the steam to the engine we have:

$$\text{Maximum Heat lost in radiation} = \frac{20{,}175}{3{,}567{,}000} = 0.565 \text{ per cent.},$$

i. e., a little over one-half of one per cent.

Since, as explained in the foot-notes, the assumed radiation data is purposely too generous, our actual radiation loss will be much smaller even than the above estimated percentage. Therefore, for commercial purposes, this loss may be entirely neglected.

In further confirmation of the above statement in the proceedings of the American Institute of Electrical Engineers, volume 25, 1906, Mr. H. G. Stott states, in referring to engine radiation losses, "This source of loss has evidently been reduced to a negligible quantity by the use of improved material and methods of heat insulation."

In the heat balance given in Mr. Stott's paper, the radiation losses are recorded as 0.2 per cent. of the heat of the coal. If the engine received 70 per cent. of the heat of the coal in the form of steam then the radiation loss amounted to 0.00286; that is, less than 3/10 of 1 per cent. of the heat entering the engine.

We have now only to consider the abstraction of heat from the steam which is converted into mechanical energy. This loss varies in inverse ratio to the efficiency of the engine. That is to say, the

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*Heat in a pound of steam at 101.3 lbs. pressure.

†A 14 x 20 engine is assumed and the outside covering of a rectangular cylinder lagging is figured as radiating surface. This surface is figured as being surrounded on its outer surface by air at 70 degrees and its inner surface to be supplied constantly with live steam at boiler pressure. Of course, the average internal temperature of the cylinder would be much lower than this, and the actual radiating surface would be much less than that assumed.

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*Coefficient of radiation assumed at 3 B. t. u. per sq. ft. per degree difference per hour. This coefficient is too high (so taken purposely) since the cylinder is usually lagged and not bare as assumed.

†339 degrees Fahr.—temperature of dry saturated steam at 101.3 pounds pressure.

more efficient the engine the less will be the heat contained in the exhaust steam mixture.

It is now only necessary to know the amount of heat in a pound of steam entering the engine and its efficiency referred to indicator horse-power to determine the heat in a pound of its exhaust steam mixture. The former quantity depends upon the temperature and pressure of the entering steam, and the efficiency upon the steam consumption of the engine.

For example, take the following data:—

Dry saturated steam at throttle, pressure ....101.3 lbs.
Temperature, Fahr. ....338.7°
Steam per Indicator Horse-power Hour ....25 lbs.
Heat in one lb. of entering steam (above 32 deg.)....1189 B.t.u.
Thermal efficiency of engine at above steam rate...........8.56%*

The heat remaining in one pound of the exhaust mixture will, therefore, be $(1.00 - 0.0856)1189 = 1085$ B. t. u. above 32°.

We may simply state that the *total heat* in a pound of exhaust *mixture* is equal to the heat in a pound of the steam at the throttle minus a percentage equal to the thermal efficiency of the engine (disregarding the negligible radiation loss). Stated as a formula we have:—

FORMULA I

$H_e = H_s \times P$ in which

H_e = the total heat above 32° in a pound of the exhaust mixture.

H_s = the total heat above 32° in a pound of the steam supplied at the throttle.

$P = 1.00$ — (the thermal efficiency of the engine referred to indicator horse-power and expressed as a decimal).

Now the form in which this heat is carried vitally affects our problem. That is, we must know what percentage of the exhaust steam mixture consists of water of condensation. After the total heat in the exhaust mixture is determined as above, its moisture content depends upon the back pressure in the engine.

To carry out our specific example let us assume the engine in question to be operating against a back pressure of 3.3 lbs. gauge measured close to the exhaust ports. The steam portion of the mixture will then contain 1154.2 B. t. u. per lb. (above 32°) and the moisture portion, which will be at a temperature of 222.4° (temper-

*Efficiency referred to indicator horse-power $= \frac{\text{output}}{\text{input}} = \frac{2545}{25 \times 1189} = 8.56\%$.

ature of steam at 3.3 lbs. gauge) will contain 190.5 B. t. u. per pound. Let x = the weight of steam in one pound of exhaust mixture. Then $1 - x$ = the weight of water in one pound of exhaust mixture. Then the heat in a pound of exhaust mixture, 1085 B. t. u. = $1154.2x + 190.5\ (1 - x)$.

Solving:

$$x = 0.927 \text{ steam (pounds).}$$

$$1 - x = 0.073 \text{ water (pounds).}$$

Hence the exhaust mixture per pound contains 0.073 pounds of moisture. The steam portion, 0.927 pounds, contains 1154.2 B. t. u. per pound, or a total of 1070 B. t. u.

If we term as *useful exhaust* that portion which consists of steam without moisture the *useful exhaust heat from our engine* will carry $\frac{1070}{1189}$ or 90 per cent. of the total heat of the steam which entered the engine in this instance.

The following formula covers the calculation for the weight of dry saturated steam contained in a pound of the exhaust mixture for all cases.

FORMULA II

$$W = \frac{H_e - H_w}{L} \text{ in which}$$

W = Weight of dry saturated steam in a pound of the exhaust mixture.

H_e = Total B. t. u. in a pound of exhaust mixture as previously defined.

H_w = The B. t. u. in a pound of water at the temperature of exhaust steam.

L = Latent heat B. t. u. of steam at the pressure of the exhaust.

The deduction of this formula, denoting the total heat of dry saturated steam at the exhaust or back pressure as H_b, is as follows:

$$W H_b + (1 - W)\ H_w = H_e$$

Simplifying we have:

$$W = \frac{H_e - H_w}{H_b - H_w}$$

Now, since $H_b - H_w = L$, we have

$$W = \frac{H_e - H_w}{L}$$

This simple method of determination may be applied to any set of engine conditions and may be relied upon for accurate results. The appended exhaust steam curves have been constructed in this manner and are here presented with a view to furnishing a convenient reference for the use of heating and ventilating engineers.

It is to be noted that the present curves are for dry saturated steam at the engine throttle, though, of course, by the same method of calculation they may be made to include both superheated steam and steam containing moisture.

From an analysis of the Formulas I and II, it will be found that the percentage of the heat of the initial steam which will appear as dry saturated steam in the exhaust (that is the result we desire which we may term R%) obeys the following laws as to its variation.

Changing the Back Pressure:

Increasing the Back Pressure Lowers the R%.

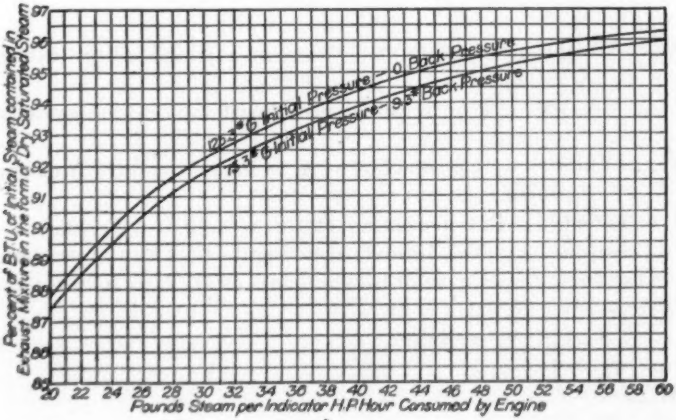
Reducing the Back Pressure Raises the R%.

Changing the Initial Pressure:

Increasing the Initial Pressure Raises the R%.

Reducing the Initial Pressure Lowers the R%.

If, therefore, we lay out one curve, representing maximum initial pressure and minimum back pressure, and another curve for mini-



mum initial pressure and maximum back pressure, all the other possible values of R% will lie between these two curves for any other combination of initial pressure and back pressure whose values lie between the selected extremes.

In order to cover practically all ranges of application, I have plotted the first or upper curve for 125 pounds (gauge) initial pressure with zero or atmospheric back pressure, and the second or lower curve for 75 pounds initial pressure with 9.3 pounds (gauge) back pressure. It will be noted that within this range any changes in initial and back pressure will affect the result no more than 6/10 of one per cent.

For applying these curves for results within this degree of accuracy, it is merely necessary to know the steam consumption of the engine per indicator horse-power per hour, and the steam pressure at the throttle. If still finer results are wanted, the back pressure should also be known, but since heating calculations are susceptible to errors which in comparison are huge, the heating engineer need only know that the back pressure does not exceed 9.3 pounds in order to get results from these curves. The results obtainable in this easy, simple manner, are, as stated, accurate to within one per cent. Absolute accuracy, if desired, can be obtained (barring the negligible quantity of radiation) by careful application of Formulas I and II herewith deduced.

It is, perhaps, hardly necessary to state that the results in the use of exhaust steam in actual practice closely agree with these determinations of heating value. Any engineer who has had the opportunity to make observations with actual measurements, has found that exhaust steam from a simple engine is worth for heating purposes nearly as much as live steam. I have myself been fortunate in having had opportunities for this kind of measurement. I might mention one case where several slide valve engines were used to furnish power whose exhaust steam was absorbed by the heating system in a large manufacturing concern. The exhaust steam just balanced the heating requirements at the time of year when I made my experiment. I shut down all the engines in the plant and supplied the heating system by using the live steam at reduced pressure. The fuel was carefully weighed and it was found that just as much was consumed during the test with the engines shut down as when all the engines were developing their full quota of power. This experiment was made for the purpose of supplying a practical demonstration which would appeal to the minds of an executive committee of a large corporation who believed that it would pay to shut down the steam engines and replace them with producer gas engines and who could not understand that exhaust steam contained within a few per cent. as much heat as live steam. I have made other tests which have checked up these results closely.

The curves presented herewith cover practically all cases of engines whose exhaust might be used for heating purposes, the range covering steam consumptions per indicator horse-power per hour from 20 pounds to 60 pounds; and in order to facilitate the use of the curves the following table of steam engine consumption is given.

Steam Consumption of Engines

For dry saturated steam, pressure range 75 lbs. to 125 lbs. gauge, back pressure range 0 to 9 pounds gauge, steam consumption of engine will vary as follows, depending upon the special combination of these factors, percentage of full load and upon the physical condition and detailed design of the engine.

Type of Engine	Steam per I. H. P. Hour—Non-Condensing	
	Range	Average
Slide valve, throttling governor, simple engine....	35 to 75	50
High speed automatic cut-off governor, simple engine	28 to 57	35
High speed, four-valve non-releasing gear, cut-off governor	25 to 45	33
Corliss, regular type releasing gear	23 to 35	30
Corliss, compound releasing gear	20 to 30	25

The steam consumption of an engine varies so widely with the variables noted, and also with the size of the engine, that this table must be used with discretion. The best way to obtain the true steam consumption of an engine is to test it by weighing the feed water to the boiler which supplied it, cutting out, of course, all other steam consumers during the test.

We have now developed sufficient information for the easy and accurate determination of the heat in the exhaust steam from engines but to make this paper more nearly complete we must add some matter relative to steam pumps, since it is from this source that a very large amount of exhaust steam may be supplied for heating. From tests which I have made, it is safe to calculate that a duplex boiler-feed-steam pump will consume from 5 to 7 per cent. of the total steam generated in the boilers which it supplies. I have further calculated that such pumps have an efficiency corresponding to about 237 pounds of steam per indicator horse-power per hour. Applying

formulas I and II, we shall find (boiler pressure 115 pounds) that R% = 99½ per cent. with 0 back pressure.

For ordinary small sized boiler feed pumps, therefore, we may safely say that the pump takes out not over one per cent. of the heat of the steam which is supplied to it and the steam consumed by such a pump amounts to approximately 6 per cent. of the steam generated by the boilers which it feeds.

Steam pumps of this type but used for other purposes will also exhaust about the same percentage of useful heat as related to the steam which enters them. Hence for the same amount of work (based on indicator horse-power hours) the ordinary steam pump consumes about eight times as much steam as a Corliss engine. Based on *brake horse-power or actual work performed,* the pump is still *more* wasteful in comparison with the engine.

These data are given merely to attract proper attention to the very large amount of heat which is made available in the form of exhaust from steam pumps. This must be computed as shown and added to the heat of the engine's exhaust in order to produce a complete estimate of the total available heat in the exhaust steam. Without such an estimate beforehand no heating work can have those qualities of certainty and efficiency which always distinguish engineering from guess-work.

DISCUSSION

During the winter, at the suggestion of this Society, experiments were made at the Sheffield Scientific School of the Yale University, under the direction of Prof. L. P. Breckenridge, by Prof. E. H. Lockwood, to determine the actual value of exhaust steam, and Prof. Lockwood has given the results of his experiments in a paper which is followed by the other discussions of the subject.

Mr. West gives some explanations of the problems in thermodynamics that enter into the proposition of the steam condensed in the running of an engine, and he follows the steam in the process through the engine until it reaches the exhaust.

At each step in the operation an explanation is given of the possible loss and tables are shown to give the losses of different types of engine, as well as charts to show the consumption at different points of cutoff explaining that the greater amount of work extracted per pound of steam, the greater would be the amount of condensation at the exhaust.

This paper further says that in passing high pressure steam from a boiler through a system of piping and thence through a reciprocating engine, a considerable heat loss is encountered, which usually results in a considerable percentage of condensation in the exhaust; with a statement that in simple engines this would run between 15 and 20 per cent.

Mr. Myers' paper states that in the running of an engine the heat of the steam is extracted in two ways; by conversion of a part of the heat into mechanical work, and by dissipation of a small amount through cylinder radiation, but he disregards the cylinder's loss as being almost a negligible factor. As an example:

Dry saturated steam at throttle, pressure............	101.3	lbs.
Temperature, Fahr.	338.7	deg.
Steam per Indicator Horsepower Hour.............	25	lbs.
Heat in one lb. of entering steam (above 32°).......1189		B.t.u.
Thermal efficiency of engine at above steam rate....	8.56%	

This may be stated by the Formula 1.

$H_e = H_s \times P$ in which

$H_e =$ the total heat above 32 deg. in a lb. of the exhaust mixture.

$H_s =$ the total heat above 32 deg. in a lb. of the steam supplied at the throttle.

$P = 1.00$—(thermal efficiency of the engine referred to indicator horsepower and expressed as a decimal).

A chart is also given showing the value of exhaust steam when supplied to the engine at different initial pressures and when emitted at various back pressures. This chart covers the limits of usual operation for engines found in heating service with steam consumptions per indicator horsepower hour ranging between 20 lbs. and 60 lbs. A subsequent calculation is made for the quality of the exhaust steam by means of the formula $W = \frac{H_e - H_w}{L}$ deduced as follows:

$W =$ Weight of dry saturated steam in a lb. of the exhaust mixture.

$H_e =$ Total B.t.u. in a lb. of exhaust mixture as previously defined.

$H_w =$ The B.t.u. in a lb. of water at the temperature of exhaust steam.

$L =$ Latent heat B.t.u. of steam at the pressure of the exhaust.

The deduction of this formula, denoting the total heat of dry saturated steam at the exhaust or back pressure as H_b, is as follows: $WH_b + (1 - W)\ H_w = H_e$. Simplifying we have:

$W = \dfrac{H_e - H_w}{H_b - H_w}$ Now, since $H_b - H_w = L$, we have $W = \dfrac{H_e - H_w}{L}$

Having found the quality of the exhaust, the total heat of the dry portion is readily computed and the values thus found are plotted on the ordinate of the chart referred to in terms of percentage of initial heat supplied to the engine. The theoretical deductions above outlined correspond to the tests made on the quality of steam from the engines of the 59th St. Interborough Power Station.

A DISCUSSION BY E. H. LOCKWOOD

Mr. Myers' paper is to be commended for the valuable information it contains regarding the amount of moisture in exhaust steam. The methods used are without doubt substantially correct, but one or two minor points are open to correction. Formula 1, can be expressed in a simpler way, by writing it

$$H_e = H_s - (2545 \div W)$$

in which

H_e = the total heat above 32° in a pound of exhaust mixture.
H_s = the total heat above 32° in a pound of dry steam at the throttle.
2545 = the heat equivalent of one horsepower, per hour.
W = weight of steam used per, for each indicated horsepower.

This formula is identical with the one suggested by Mr. Myers, but is written in a different form to avoid the term "thermal efficiency," which serves no useful purpose. When interpreted in words, the formula 1 reads thus:

The heat in the exhaust mixture (H_e) is equal to the heat originally in the steam entering the cylinder, (H_s), minus the heat turned into work, (2545 ÷ W).*

The identity of the formula given above with the one in Mr. Myers' paper can be shown by solving the numerical example where H_s = 1189, W = 25, whence,

$$H_e = 1189 - (2545 \div 25) = 1087 \text{ B.t.u.}$$

The value found by Mr. Myers for the same problem was 1085 B.t.u., the difference being due to omitted decimals in calculating.

A slip was made by Mr. Myers in the calculation of the useful heat in the exhaust mixture, but, fortunately, the error is small enough to be neglected. The useful heat in the exhaust mixture cannot be found correctly by multiplying the total heat of dry steam at the exhaust pressure by the dryness factor. The error

*In his paper on Engine Condensation, Mr. Perry West presents a formula for the condensation due to work in the cylinder, which differs from the one just stated, by including the work of admission in addition to the indicated horsepower. I believe Mr. West has fallen into error in this instance.

is shown in the example worked out by Mr. Myers, where, starting with 1085 B.t.u. in a pound of exhaust mixture of dryness 0.927 the "useful" heat is found to be only 1070 B.t.u., a decrease of over one per cent., for which there is no justification. Strictly speaking, the useful heat in the exhaust mixture is neither of the figures mentioned, but only the heat given out in condensing the steam portion of the mixture, which can be calculated by multiplying the latent heat by the dryness factor. This error, as noted above, is small enough to be neglected in practical work, but happily, there is another factor which tends to offset it, namely, the omission of radiation loss from the cylinder. The reasons for neglecting the radiation loss from the cylinder have been mentioned, but I believe this loss is greater than was indicated and may frequently be as much as one or two per cent. Thus these two losses in opposite directions may neutralize each other and leave the final result unaffected.

The correctness of Mr. Myers' results have been verified by experiments made at the Mason Laboratory of Mechanical Engineering, Sheffield Scientific School, during the past year. As little experimental work has been published on moisture in exhaust steam, it may be of interest to give a summary of these tests.

The apparatus consisted of a 25 horsepower Corliss engine, cylinder 10 x 24, speed 75 r.p.m., exhausting through a 4 inch pipe into a Wainwright surface condenser containing 50 square feet of cooling surface, and having appliances for accurate measurement of condensate and cooling water. A 3-inch Cochrane vertical separator furnished practically dry steam at the throttle.

The heat contained in a pound of the exhaust mixture, denoted by H_e, was determined by direct measurement of four quantities:

(a) Weight of condensate.

(b) Heat contained in the condensate above 32 deg. F.

(c) The heat absorbed by the cooling water.

(d) The heat radiated from the condenser shell and piping.

To determine H_e, items (b), (c), (d) were added and the sum was divided by item (a). In making the experiments the weights and temperatures were taken with the greatest care possible. The condensate was not all weighed at the condenser outlet, but part was dripped from the exhaust pipe between the cylinder and condenser. In calculating the heat in the condensate, allowance was made for the difference in temperature of this drip and the condenser discharge. About three-fourths of the moisture in the exhaust pipe was removed in this way, and the remainder was carried over into the condenser. Item (d) was measured by a separate

test, filling the condenser and piping with a supply of dry steam at atmospheric pressure and measuring the weight of steam condensed per hour. The condensation factor obtained in this way was about 3 B.t.u. per sq. ft., per deg. difference of temperature, per hour.

Line 16 contains the heat in one pound of exhaust mixture as experimentally determined, while line 17 gives the ratio of this value to the total heat in the initial steam. A direct comparison can be made of the percentages in line 17 with the corresponding figures taken from the diagram of Mr. Myers' paper.

The average experimental result in the Corliss engine tests is 92 per cent., while the value taken from Mr. Myers' diagram for the same conditions is 93 per cent., showing a satisfactory agreement. Line 19 contains the percentage of moisture in the exhaust mixture which averages about six per cent. for the three runs.

TESTS FOR MOISTURE IN EXHAUST STEAM

From a 10 x 20 Corliss Engine at the Mason Laboratory, Sheffield Scientific School, Yale University

1. Date of test, 1914.......		Dec. 19	Nov. 3	Dec. 19
2. Speed	r.p.m	74.2	73.5	72.6
3. Steam press. at throttle	gauge	82.	70.	81.
4. Press. in condenser....	gauge	0	0	0
5. Indicated horsepower...	i.h.p.	17.2	21.0	26.6
WEIGHTS AND TEMPERATURES				
6. Weight of condensate..	pounds /hr.	622.1	728.	840.
7. Water rate of engine..	per i.h.p.	36.2	34.6	31.6
8. Weight cooling water..	pounds /hr.	6090.	13,200.	8,460.
9. Temp. cooling water inlet	deg. F.	75.7	115.	75.
10. Temp. cool'g water outlet	deg. F.	171.6	167.	170.3
11. Temp. condensate.......	deg. F.	130.8	145.	138.5
HEAT AND MOISTURE RESULTS				
12. Heat in condensate......	B.t.u. /hr.	64,900.	84,700.	91,600.
13. Heat in cooling water...	B.t.u. /hr.	584,000.	686,000.	811,000.
14. Heat radiated from cond.	B.t.u. /hr.	25,400.	25,400.	25,400.
15. Total heat acctd. for...	B.t.u. /hr.	674,300.	796,100.	928,000.
16. H_e (Heat in one pound of exhaust mixture)..	B.t.u.	1,084.	1,091.	1,102.
17. Per cent. of heat of initial steam contained in exhaust mixture.....	per cent.	91.3	92.	93.
18. Dryness of exhaust mixture	per cent.	93.2	93.9	95.
19. Moisture in exhaust mixture	per cent.	6.8	6.1	5.0

The radiation loss from this cylinder was carefully measured to determine whether the amount of moisture found by the experimental tests could be calculated in advance from the known indicated water rate and estimated heat loss from the cylinder. It

was found that the amount of moisture calculated in this way agreed very closely with the experimental values. In measuring the heat radiating surface it was decided to subdivide it into three parts, each having a different average temperature. First, the bare metal containing high pressure steam, the throttle valve, flanges, etc., a total of 3.2 sq. feet at 320 deg. F. Second, the bare metal parts receiving heat from the cylinder but not lagged, such as the valve bonnets, valve stem and arms, wrist plate bracket, cylinder head cover, and portion of bed receiving heat by conduction, the total being 18.8 sq. feet at an assumed temperature of 200 deg. F. Third the lagged sides and the cylinder feet a total of 11.9 sq. feet, assumed at a temperature of 140 deg. F. A heat loss co-efficient of 3.5 B.t.u. per degree, per sq. foot, per hour was chosen, a large value, owing to the exposed position of the cylinder and free circulation of air about it.

When the heat loss was calculated from these figures and compared with the heat in the steam passing through the cylinder, it was found that it amounted to 2.1 per cent. at the lighter load and 1.5 per cent. at the largest load, the exact values being given in the table below. These values for radiation from the cylinder are larger than those given in Mr. Myers' typical example, partly because the lagging was defective, and partly because a small cylinder always has more surface in proportion to its volume, than a larger cylinder would have.

CALCULATED MOISTURE IN EXHAUST

From 25 H.P. Corliss Engine

Based on radiation from cylinder and heat converted into work, assuming a perfectly dry steam supply.

1. Indicated horsepower	17.2	21.0	26.6
2. Water rate per i.h.p.	36.2	34.6	31.6
3. Moisture produced by work, per cent.	3.5	4.1	4.6
4. Moisture produced by radiation, per cent.	2.1	1.8	1.5
5. Total calculated moisture, per cent.	5.6	5.9	6.1

COMPARISON OF CALCULATED MOISTURE WITH EXPERIMENTAL MOISTURE

Calculated moisture, average for three runs......... 6 per cent.
Experimental moisture, average for three runs...... 5.9 per cent.

From the foregoing considerations it may be safely assumed that the percentage of moisture in the exhaust steam from any

engine can be calculated quite closely, provided we know the i.h.p., the water rate, and the cylinder dimensions, lagging, etc.

Prof. Kent: I think Mr. West is right in his general proposition, but some of his figures appear to be too high. He says, 'Considerable heat loss is encountered which usually results in a considerable percentage of condensation in the exhaust. I should say with a simple engine this would run between 15% and 20%.' There may be this condensation in the forward stroke of an engine, but most of it is re-evaporated in the exhaust stroke, and the exhaust steam may contain 85 per cent. of the heat of the steam delivered to the engine.

Mr. David Moffat Myers: Mr. Perry West's paper on engine condensation is interesting in its analysis of the action of steam before entering the engine cylinder and throughout the stages of its expansion in the cylinder. Regarding, however, any direct application of the data presented in this paper for the calculation of the heating values of exhaust steam, it seems to me that Mr. West has attacked the problem in a very roundabout way and his conclusion that the percentage of condensation in the exhaust of simple engines would run between 15 and 20 per cent. is inadequate, and I regret to state very inaccurate, as will be developed in the following discussion.

In the first place, while of course the heating engineer is interested to some extent in the amount of moisture in exhaust steam, he is far more interested in the *heating value* of this steam, and Mr. West's paper fails to give him any simple data from which to obtain this figure. In Formula II. of my paper, W is the weight of dry steam in the exhaust per pound of exhaust mixture or per pound of entering steam. Therefore 1 — W gives the amount of moisture in the exhaust. Solving for the moisture under various extreme conditions of operation gives the following tabulated results:

FOR DRY SATURATED STEAM AT ENGINE THROTTLE

Steam Gauge Pressure at Throttle	Back Pressure Gauge	Steam Consumption of engine per i.h.p. Hour	Quality W = Weight, lb., of dry saturated steam in lb. of exhaust mixture	Moisture in exhaust mixture 1 — W
125.3 lb.		20 lb. 60 lb.	0.911 0.998	0.089 = 8.9% 0.002 = 0.2%
75.3 lb.		20 lb. 60 lb.	0.892 0.981	0.108 = 10.8% 0.019 = 1.9%

Thus, between the selected extremes of non-condensing operation the moisture in the exhaust steam will vary between 0.2 per cent. and 10.8 per cent. and not between 15 and 20 per cent., as stated in Mr. West's paper. I take it that Mr. West has included in this estimate the moisture entrained with the boiler steam as well as the slight condensation of the steam due to radiation from the piping to the engine. But, except in very rare instances, the amount of moisture in the steam at the throttle of an engine would not exceed 3 per cent. By altering my results to include these items of condensation, *i.e.* by adding this 3 per cent. the exhaust steam would contain approximately from 3.2 per cent. to 13.8 per cent. of moisture as compared to perfectly dry saturated steam. Thus my highest figure representing the extreme conditions favorable to condensation is less than Mr. West's lowest estimate.

I have applied my simple formula, which is at the same time, scientifically correct (barring the negligible element of cylinder radiation), to about 27 different sets of conditions. As deduced in my paper the percentage of available heat in the exhaust steam varies with three factors: Engine Efficiency, Initial Steam Pressure, and Back Pressure. I have also shown that this percentage is reduced when the back pressure is increased and that it is increased when the initial pressure is increased. I have also indicated the fact that the higher the efficiency of the engine the more will be the moisture and the less the percentage of available heat in the exhaust steam.

All three of these variable factors are taken care of in my curves as shown, so that all the heating engineer has to know is what kind of an engine he is using, the steam pressure and the back pressure. In fact, for all practical purposes, all he really needs to know is merely what kind of an engine he is using in order to get the desired results directly from the curves. My results are given in terms of *useful heat in the exhaust* and *not* moisture.

To illustrate with a practical example, suppose a good Corliss engine is taking 100 boiler horsepower per hour of steam. From my table we see that such an engine will take about 30 pounds of steam per indicator horsepower per hour. Applying this figure to the curves on the chart we see that the useful heat in the exhaust from this engine is 91¾% to 92½% of the initial heat supplied. That is to say, the engine in this case will supply exhaust steam of equal heating value to 92 boiler horsepower per hour. In other words, the engine has reduced the heating value of the boiler steam by 8 per cent.

I have not had time to work out Mr. West's four formulas to see how his values would compare, and I fear that very few persons would find time to do so. If they did, there would be too many chances to make mistakes. Furthermore, to use these formulas the engineer will have to take indicator cards from his engine in order to obtain the indicator horsepower and the (P_1) "average difference in pressure between admission and compression." He would then have to look up the latent heat of steam at exhaust pressure, the total heat of steam at admission pressure and the total heat of steam at exhaust pressure. In addition to all this he would have to calculate Q_1, the cu, ft. of steam per hour at admission pressure, *and all this before he could use the single formula for C_2.* The other three formulas add still further to the complexity and difficulty of the computation, so that, although apparently based upon sound theory the formulas deduced are too unwieldy for practical purposes, even if correct, for which latter feature I am not prepared to vouch.

Mr. West gives no specific results to cover the usual range of practice, but it is his opinion that with simple engines the condensation would run between 15 and 20 per cent. Assuming that dry saturated steam enters the engine, my own calculations prove that engines operating on steam with initial pressure anywhere between 75 and 125 lbs. gauge and against a back pressure of anywhere between 0 and 9.3 lbs. gauge will emit an exhaust mixture containing between 0.2 per cent. and 10.8 per cent of water. These figures are widely different from the 15 and 20 per cent. quoted by Mr. West, and in order to demonstrate beyond possible doubt the accuracy of my formula *for the benefit of those who do not rely upon scientific deduction,* I have compared its results with the actual test results of moisture in exhaust steam by Messrs. H. G. Stott and R. J. S. Pigott. I refer to their paper before the American Society of Mechanical Engineers, Trans. 1910, which gives the data of elaborate tests on the steam engine turbine units at the 59th Street Interboro Central Station. Tests were made on the quality of the initial and exhaust steam and checked in a most thorough manner. There has probably never been made in the history of engineering so complete and accurate a series of engine tests as these to which I refer.

The set of parallel data for the present purpose I have taken from Test No. 62, of Series E. and F. These data are as follows:

Steam pressure at throttle	195 lb. absolute.
Quality of steam	99 per cent.

Combined thermal efficiency of engine and generator efficiency .... 10.7 per cent.

Indicator Horsepower per Kilowatt 1.456

Electrical and mechanical efficiency $\frac{1.34}{1.456}$ ...... 92 per cent.

Thermal efficiency referred to indicator horsepower $\frac{10.7}{0.92}$ .... 11.63 per cent.

Quality of steam found in exhaust from low pressure cylinder .... 89.8 per cent.

89.8 per cent. — 1.8 per cent. = 88.0 per cent.

Water removed between high pressure and low pressure cylinder, 4057 lb. per hour.

Total water to engine, 226,365 lb. per hour.

Correction (approximate) for this water $\frac{4057}{226365}$ = 1.8 per cent.

Corrected quality as found .... 88.0 per cent.

The application of my Formulas I. and II. gives a quality (W) of 88.2 per cent. when a cylinder radiation loss of 1 per cent. is added to the thermal efficiency in Formula I. The small discrepancy of two-tenths of one per cent. may be accounted for by Formula I. $H_e = [1.00 - (0.1163 + 0.01)] \times 1190 =$ 1040 B.t.u.

Formula II. $W = \frac{H_e - H_w}{L} = \frac{1040 - 190.5}{963.7} = 88.2$ per cent.

We have now compared the values obtained by my formulas and curves with those resulting from actual tests made on the exhaust steam from engines, and we have seen that they coincide.

In corroboration of the theoretical determinations I would refer to some special cases relating to the heating value of exhaust steam, which are worked out by Mr. James D. Hoffman, M. E., Past President of this Society, as given in his Hand Book for Heating and Ventilating Engineers. I have taken two of these cases as follows:

First case—simple high speed engine. Quality of steam 98 per cent; initial pressure 100 lbs.; steam consumption per indicator horsepower per hour, 34 lbs.; back pressure, 2 lbs. gauge. Mr. Hoffman gives for these conditions as the heating value of exhaust steam 94 per cent. of the heat value of an equal weight of dry saturated steam under 2 lb. gauge pressure, that is 94 per cent. of 1153 B.t.u. or 1084 B.t.u.

From my curve it will be seen that under the above stated conditions my result would be 93 per cent. of the heat of a lb. of dry saturated steam at 100 lb. pressure. Taking 93 per cent. of 1189 B.t.u. and correcting for the quality of the initial steam will give 1083 B.t.u., that is the same result within slide rule accuracy.

Consider now the second case, Mr. Hoffman has worked out, that of a compound Corliss engine working under 125 lb. initial pressure; quality of steam 98 per cent; with steam consumption of 22 lb. per indicator horsepower per hour and a back pressure of 2 lb. gauge. The heat value of the resulting exhaust steam is given as 90 per cent. of the heat in the same weight of dry saturated steam at 2 lb. pressure, or $0.90 \times 1153 = 1038$ B.t.u.

Taking the value for an engine of this consumption from my curve we get 89 per cent. of the value of *initial* steam when same is dry saturated at 125 lb. pressure. That is, 89 per cent. of 1192 B.t.u. equals 1061 B.t.u. and correcting for the quality of initial steam gives 1041 B.t.u. for my value on Mr. Hoffman's basis. Thus it will be seen that the value for my curve compares closely with that obtained by Mr. Hoffman.

I trust that I have now been able to show conclusively that the formulas and curves which I have worked out to give the heating value of exhaust steam from any kind of an engine are accurate and consequently give values which can be relied upon in practical work. The deduction which I have used is correct scientifically and compares accurately with practice as shown by comparison with moisture tests made on exhaust steam from engines working under known conditions.

I may further add that the radiation loss from cylinders of the same dimensions is practically constant. Consequently the *percentage* of radiation loss increases with the efficiency of the engine. That is to say, the less the heat supplied the greater will be the radiation loss compared to this quantity. Therefore, since (generally considered) engines whose exhaust is used for heating belong to the less efficient class, the error introduced by disregarding this loss is indeed negligible in its effect upon the computation of the heating value of exhaust steam.

Mr. West: Mr. Myers seems to have made quite an arraignment of my paper, but I understand him, to sum it up in a few words, as being "unwieldy and hard to manage."

I have no complaint to make as to this criticism, as it was not my intention to present something simple and easy for any one to handle, at the expense of a thorough and scientific discussion of the subject. I stand on my discussion to the effect that it is the

theoretical and proper way to get at this subject of engine condensation.

Mr. Myers' formula is based on the theory that the heat extracted from the steam is exactly equal to the work done by the engine and that the condensation from radiation is practically negligible. My theory on the other hand is correct for application to engines running under all possible conditions. His formula does not take into account the fact that the radiation element might be quite considerable under certain conditions, nor does he take into account the factor of different points of cut-off in the estimating of condensation due to work in the cylinder. Where an engine is cutting off practically at full stroke, for instance, there will be practically no condensation due to the work in the cylinder, because there is no work being done by expansion. This fact is not taken into account in Mr. Myers' theories.

In regard to radiation, Mr. Myers has taken as a sample, the subway engines, which are very large double engines, and, of course, as we all know, the smaller the engine the greater the percentage of radiation, other things being equal. My experience in figuring this out is that the total condensation from boilers to exhaust amounts to about 15 to 20 per cent. for small plants.

As I have said, my discussion of this subject was not intended to be primarily simple, but to be of a more thorough and fundamental nature, so that any one after understanding same, might be able to go to the bottom of this subject for himself. I have not attempted to make up curves or tables for quick application, regardless of conditions, but rather to furnish data for the proper and scientific solution of any engine condensation problem under various conditions.

In elucidation of my theory, the diagram, Fig. A, shows an

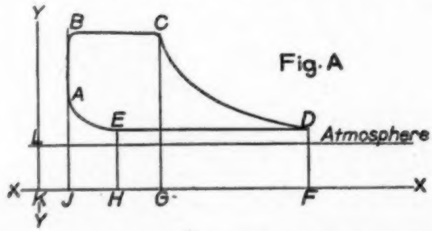


Fig. A

ordinary indicator card. Steam is admitted along the line BC and performs an amount of work on the piston represented by the area B C G J. This work is not done at the expense of condensation in the steam, since it is the regular work of displacement

performed in the boiler and simply transferred to the piston. This is the same work that is performed every time steam is generated under pressure, and does not cause condensation. After cut off at C, the steam expands along the line C D and performs an amount of work on the piston represented by the area C D F G. This work unlike the work of admission, is performed at the expense of the heat and condensation in the steam itself and causes a corresponding amount of heat to be extracted. After release at D, a certain amount of negative work is performed by the piston. This work is not performed upon the steam, however, as it neither expands nor compresses same, but is performed on other matter through the steam.

Up to the beginning of compression at E, this negative work neither adds to nor subtracts heat from the steam, since there is no work being done upon the steam itself.

From E to A the steam remaining in the cylinder is compressed and an amount of heat is thereby added to same equivalent to the work represented by the area A E H J. We see, therefore, that the only factors of work which are really connected with the condensation in an engine cylinder are those of expansion and compression. In the formulæ the work of compression has been neglected on account of the very small bearing which it would have on the results.

From this it is readily seen that the point of cut off in the cylinder and consequently the amount of work of admission has a marked effect upon the amount of condensation due to work in the cylinder.

For instance, in the case of an engine or pump cutting off at, or near, full stroke such as shown by this diagram, Fig. B, almost the entire work would be that of admission and would cause practically no condensation at exhaust. The only condensation would be caused by the expansion from C to D. In the practical condensation of this exhaust under a constant pressure such as in a heating system, the steam will expand down to this pressure after release as indicated by the figure L G H M.

So that the volume and pressure so represented correspond to the volume and pressure of the steam remaining. In so expanding the steam performs an additional amount of work represented by the area E G H O. The quantity of heat represented by this area must, of course, be deducted from the heat in the exhaust steam. In the first case, therefore, we should only deduct for the work represented by the area C D F G, and in the second case for the area C D O N and E G H O.

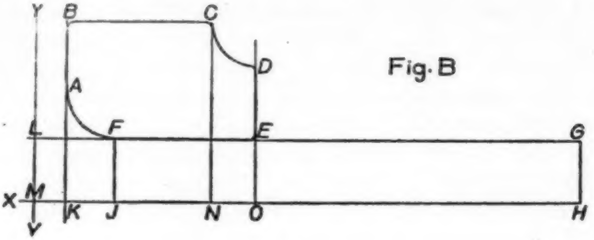


Fig. B

In a great many cases it will be found, of course, that these areas are so nearly the same as the indicated horsepower area that the method used by Mr. Myers is quite accurate enough for practical purposes.

As a matter of fact, I have used this same method very often.

Mr. J. S. Otis: I cannot conceive how it is possible, under ordinary conditions, to get a 20 per cent. loss from steam passing through a non-condensing engine. I have never found such a condition in practical experience, and I would like to know under what conditions it was obtained. Take, for instance, a high speed engine of 100 H.P. having a 13" x 12" cylinder and running at approximately 375 revolutions per minute. Under full load it would cut off at one-quarter. The cubical contents of the cylinder itself is 92/100 of a foot. If we cut off at one-quarter we would have 23/100 of a foot. This is the equivalent of 6/100 of a lb. at 100 lbs. pressure. We cannot expand the steam unless we drop the pressure, consequently, we must expand to the full size of 92/100 of the foot. Then it has dropped down to 10½ lbs. pressure. The point where expansion is stopped is the point where the steam pressure will not overcome the frictional resistance. Consequently, a high speed engine will allow for drop in steam pressure to about 10½ lbs. under full load; so that the steam as it leaves the cylinder is at 10½ lbs.

Now suppose we drop it to three lbs. pressure in the exhaust, you will find by the B.t.u. left in the steam at 3 lbs. that the total drop has only been 8.7 per cent. of the value of the steam. This, of course, is theoretical. Mr. Myers, in his paper, finds the actual drop to be about 10 per cent. This is easily accounted for in loss from exposure and frictional resistance in the port passages. But I cannot understand how this percentage can be increased to 15 or 20 per cent. This seems unreasonable, and I believe we should have a more explicit explanation as to how and under what conditions these results were obtained.

Mr. Carrier: I think this paper of Mr. West throws a great deal of light on the subject and I believe it is enlightening to all of us. Regarding the formulas, perhaps I agree with Mr. Myers, and I would even go further, I don't see why formulas are necessary. Simply a little common sense arithmetic. You have a given number of pounds of water which is evaporated and converted into steam at a definite pressure, and it requires a certain amount of heat to accomplish this, as Mr. West says. Such data are given in various tables. If work is taken out of the steam the value of the work is lost to the steam and if we subtract that difference and the slight amount due to radiation, we have the remainder that is available for heating.

Mr. West: I take a decided issue with Mr. Carrier. When putting steam through a cylinder, heat is taken out, and that must be subtracted from the initial temperature. There are only two ways in which heat can be taken out of steam when passing through an engine to the exhaust pipe. One way is cylinder radiation. This is an extremely small quantity and is around 1 per cent. The cylinder loss is much larger in proportion on a large engine than on a smaller heating engine, because it takes more work from the steam. You put less work per horsepower in the big engine than in the little engine. Where initial supply is large per unit horsepower you have a big divisor, consequently, a smaller percentage. The fact is, the smaller the engine the less the percentage of radiation loss.

Suppose you change to the Corliss engine, the radiation is practically the same as before, but since you use less steam, your radiation loss will be, naturally, greater; the more efficient the engine the greater the radiation loss, consequently, if you start with the greatest radiation loss which you get with the most efficient engine, you are safe in eliminating and neglecting that loss entirely in small engines as used in office buildings and factories."

Mr. Carrier: I think I understand Mr. Myer's analysis, but to be sure I have just worked out my understanding of it and would like to have him verify it or otherwise. Suppose, for example, we have an engine using 30 pounds steam per hour: This means one-half pound per minute, the total heat of that amount of steam would be 590 heat units. We would have to subtract 42½ B.t.u. which leaves us 547 heat units available at exhaust at atmospheric pressure. We have an amount of steam containing only 547 heat units and if we condense it at that pressure, we would have 547 heat units, which means about 7 per cent. of heat that has disappeared in work and radiation. I think that is his analysis of it.

Mr. Myers: Just one thing, Mr. Carrier, enters into this calculation. Heat remaining in the steam would be according to the theory which you state, but that would be in the form of steam plus water, and if you take the water out, you will get better results.

Mr. Carrier: I said it was an interesting analysis, and I thought it could be expressed even more simply than Mr. Myers expressed it by the steam tables.

Mr. Ripley: I have measured the heat in exhaust steam under different pressures, and with varying loads on two different engines. My observations agree very closely with Mr. Myers' theory. I have gone a step further and have found conditions under which exhaust steam, pound for pound, is superior to live steam or boiler steam. I have averaged twenty-one observations on five different days and these show that on the average exhaust steam was 8 per cent. better for heating purposes than boiler steam, and that it took longer to deliver the same amount of heat by boiler steam than it did with exhaust steam.

I obtained these results in a plant using superheat steam and with engine running below its rated capacity. Some observations show as high as 14 per cent. superiority. If you examine this steam you will find that which has gone through the engine is drier than the steam coming out of a reducing valve. I had only five feet of pipe between the engine and outlet and only five feet between reducing valve and outlet, so that the question of steam giving up heat through iron is eliminated. We have here pure scientific phenomena.

Mr. Blackmore: In regard to the question of exhaust steam being of greater value than live steam, I referred this question to Prof. Allen as one of the things that ought to be thrashed out, and he said, so far it had not been proved. He also said it would be worth while to make some actual tests to see if there is any difference. He is going to make three tests, one of high pressure and another with steam through a reducing pressure valve with from two to five pounds pressure, and with exhaust steam at the corresponding pressures to see if he can discover any difference in value. It is possible that we may get the results of these tests at the Summer meeting.

Mr. Carrier: Mr. President, I am glad the question of live and exhaust steam has been raised. I have had occasion to try that question out. I have made a good many radiator tests, and I find that it depends largely on the flow through the radiators. If the flow is rapid, condensation has not time to occur. If it is slow, then the statement as made holds true, because then the condensa-

tion becomes moist so soon after entering, that it only affects a small portion of the radiator. We all know that we can't produce work without absorbing energy, and it must come from somewhere. Explanation of this is given in the steam tables.

Mr. J. A. Donnelly: I ran quite a number of tests on radiation with exhaust and live steam to see if I could find any difference in the amount of water condensed. I found substantially the same condensation. As far as I could find out we get the same condensation with either exhaust or live steam

Mr. Perry West: I want to say this, regarding what Mr. Carrier has just said, if Mr. Carrier has a more simple means of getting the results, I should like to become familiar with them.

Mr. Barron: I am anxious that Mr. West should answer this question of mine, if he will kindly do it. In his small indicator card showing steam going through a cylinder, he made the statement that you get the energy out of the engine by the displacing of the steam not by the transformation of energy. I would like to have him explain what he means when he said that the B.t.u. is not transformed, although work is being done?

Mr. Perry West: Mr. Barron's question, I think, is quite easy to answer. The work is put into the steam in the way of, what we call, the external work of generation. In generating steam under pressure there are two factors, or rather three factors, that enter into the amount of heat that must be put into that steam.

One is the B.t.u. to raise the temperature of the water to the temperature of evaporation, another is what we call the initial latent heat of evaporation, and the third is what we call the latent heat necessary to overcome the external work of displacement. This last factor is in the form of work and does not cause condensation when expended on the piston.

Mr. Quay: Mr. Myers spoke about steam pumps requiring a large amount of steam per horsepower. Different types of steam pumps use different amounts of steam. Our members might be interested enough to have our Committee on Tests give some information in regard to the amount of steam required by the different types of boiler feed and other steam pumps per horsepower.

Mr. Myers: I want to call attention to the fact that I attempted to deal with this problem in the simplest possible way, and I think I have done so. Whether the final results would agree with Mr. West, I do not know. If Mr. West would care to work with me, I would be very glad to compare results with him. As far as being practical or not, I have shown conclusively that my results are practical as well as scientifically correct.

Mr. Perry West: Mr. President, I want to, if I may, correct what I assume, from the discussion here, to be a false impression. That is, that the results as figured out by my formulas would differ greatly from the results as figured out by Mr. Myers' formulas. I doubt if there would be any great difference. As I stated when I was delivering my paper, I am inclined to believe that in practical application they would work out almost the same. However, in my formulas I clearly give the theoretical and correct value under any conditions. In other words, I imagine Mr. Myers' figures will give results which are correct in practical cases. I want to add further that I feel perfectly satisfied that the theory which I present is correct and accurate as against Mr. Myers' theory. I think the subject is really of enough importance to be thrashed out and settled one way or the other. I am quite sure that the gentlemen who have discussed this, as well as Mr. Myers, will agree with me when they go into the subject.

Pres. Lewis: We have heard for years, operating engineers make the claim that they could run their plants for heating more economically through the week when they had exhaust steam, than on Sundays, when they did not have exhaust steam. Text-book data to the contrary, there must be some cause for such statements.

CCCLXVI

THE RECIRCULATING OF AIR IN A SCHOOLROOM IN MINNEAPOLIS

BY FREDERIC BASS

The experiments to be described below were conducted in the same school buildings of Minneapolis, the Jackson and the Adams, in which similar experiments were conducted the year before and which were described in Vol. 19 transactions of this Society on pages 328-350. The period covered by the experiments was approximately four months.

The experiments this year were carried on in co-operation with the New York State Commission on Ventilation, and the funds were provided from a gift made to that Commission by Mrs. Elizabeth Milbank Anderson through the New York Association for Improving the Condition of the Poor.

One room in each of the school buildings named was selected for experiment. In each room the pupils were of the same grade and about nine years of age. The room in which the air was recirculated had an average attendance of 39.5 and the average quantity of air furnished per pupil was 8.9 cubic feet per minute. The air was introduced at each desk top through funnel shaped orifices and also through openings in pipes running along the top of the black boards at the end of the room. Air was exhausted at the ceiling through fifteen 3-inch orifices, evenly spaced. The air exhausted from the room was for the greater part of the time passed through a Warren Webster air washer, cooled about 15 degrees Fahrenheit and returned to the room.

In the Adams School the usual system of plenum fan ventilation was used. The average attendance in this room was 41 and the amount of air furnished per pupil was 35.4 cubic feet per minute. The air was delivered by the fan supplying the whole building, was from the outside and received no treatment except that of being heated.

RECORD OF ROOM CONDITIONS

Temperature records were kept by recording thermometers attached to inside walls about 7 feet above the floor. The temperature of the air in the room was also ascertained at ten different points, five evenly spaced at the floor level and five at the height of about four feet above the floor. The temperature of the incoming and outgoing air was also ascertained. Average temperature at inlet 56.0; average temperature at outlet 70.4.

The relative humidity of the air was determined by use of hygrodeiks which had been compared with a sling psychrometer.

The carbon dioxide contents of the air of the rooms were determined by the use of a Petterson-Palmquist apparatus.

Dust counts were made by means of filtering three cubic feet through 25 grams of resorcin. The average results are shown in Table I.

TABLE I—AVERAGE CONDITIONS OF ROOMS

	Temp.	Rel. Humidity	CO_2 p.p. 10,000	Dust Count
Adams (control)......	67.2° F.	42.2%	9.1	225,000
Jackson (experimental).	65.3° F.	46.3%	12.5	105,000

Outside temperature, humidity, precipitation, direction and velocity of the wind, were also recorded.

It was impossible to keep the windows closed at all times, and as it was not practical to keep an observer at the school throughout each day the periods in which the windows were opened were not at all times determined, although it is certain that windows were closed in the experimental room in the Jackson School except for very short periods in the later days.

PHYSIOLOGICAL EXAMINATION OF PUPILS

It was found necessary to get the written consent of the parents of all the pupils before the school board would allow the work to begin. The consent of the majority of the parents was obtained with some difficulty. At the beginning of the experiment, at the end and at a number of times during the experiment the children were all examined by a physician. They were examined for physical defects at first. Blood pressure was determined in two positions, but the technique followed by the physician was not such as to give the best results. Records of attendance, of progress in

height, weight and chest measurements were also observed. Table No. II shows the result of the examinations.

TABLE II—GAIN IN MEASUREMENTS OF PUPILS AND ATTENDANCE

	Chest	Height	Weight	% of Absences
Adams	0.14 in.	0.409 in.	1.28 lbs.	5.3%
Jackson	0.58 in.	0.298 in.	0.75 lbs.	4.1%

At the time when the consent of the parents to the examination of the children was obtained, careful examination of the home environment of each child was estimated. They are classified in Table No. III.

TABLE III—HOME CONDITIONS

	Good	Fair	Poor
Adams	12	3	3
Jackson	8	3	7

The children of the control group not only lived in better houses but they were reported by the physician as giving evidence of better care, their teeth being in better condition, their bodies cleaner, etc.

PSYCHOLOGICAL TESTS

Four psychological tests were given, a division test, a substitution test and two cancellation tests. The blanks used in these tests are shown on Plates 1, 2 and 3. The results from these tests have been plotted and are shown on Plates 4 to 7 inclusive.

CONCLUSION

An examination of the conditions in the rooms indicates that the temperature in the experimental room was 1.9 degrees lower than in the control room. It was also noticed that the temperature at the floor level averaged only 2.4 degrees lower than that four feet above the floor. This seems to indicate that upward ventilation does not leave the floor undesirably cold. Numerous examinations of air currents below the level of the currents on the desks showed that the air was continually in motion. The relative humidity of the two rooms differed by 4.1 degrees. The carbon dioxide of the experimental school was, as might be expected, considerably higher than that of the control. The dust count in the control school was, as might be expected, greater than that in the experimental school where air washing was used. The experimental group gained less

in height but increased more in chest measure. The absences due to sickness were greater in the control than in the experimental group, the difference not being very great in any of these particulars.

A	B	C	D	E	F	G	H	I	J	K	L	M	N
17	34	23	16	32	35	21	30	19	20	29	13	24	31

O	P	Q	R	S	T	U	V	W	X	Y	Z
11	25	12	26	33	15	28	36	18	27	22	14

28	19	17	24	31	18	16	30	21	14
29	12	25	33	35	15	22	26	20	27
34	11	23	13	36	32	20	11	28	30
16	22	12	19	23	18	17	29	34	33
15	31	13	26	25	27	24	14	21	36
35	33	20	32	30	19	26	23	33	18
16	14	12	21	17	25	31	11	22	34
35	29	15	36	24	28	27	13	30	33
15	13	27	36	11	12	34	14	17	32
22	35	18	16	19	25	29	20	24	28
31	26	23	21	28	34	18	30	20	29
36	21	14	16	23	22	11	31	24	33
25	17	27	32	12	13	15	35	19	26
23	25	32	14	34	31	36	24	13	11
22	27	21	17	33	35	30	20	28	15
29	19	16	12	26	18	28	19	17	24
31	18	16	30	21	14	29	12	25	33

PLATE 1

In the division test the experimental group showed a lower standard of accomplishment but at the same time greater absolute progress and consequently considerably greater relative progress than the control group. In the substitution test a higher standard was again indicated for the control group but the progress of each group was about the same. In the cancellation "B" test the control group overtook the experimental group but the rate of progress was not

6)210	7)441	8)456	9)351	6)486	7)518	8)568
7)357	8)648	9)162	6)534	7)483	8)416	9)414
8)184	9)864	6)306	7)609	8)680	9)198	6)294
9)108	6)474	7)497	8)536	9)432	6)582	7)413
6)102	7)301	8)200	9)657	6)258	7)469	8)592
7)147	8)584	9)117	6)558	7)294	8)192	9)342
8)264	9)846	6)318	7)273	8)664	9)405	6)426
9)774	6)588	7)168	8)776	9)387	6)336	7)588
6)204	7)259	8)688	9)324	6)228	7)112	8)360
7)336	8)600	9)468	6)366	7)476	8)656	9)837
8)376	9)243	6)576	7)252	8)128	9)693	6)234
9)261	6)492	7)126	8)784	9)279	6)222	7)616
6)321	7)329	8)144	9)828	6)174	7)672	8)520
7)665	8)321	9)666	6)330	7)287	8)448	9)216
8)440	9)567	6)378	7)623	8)696	9)369	6)444
9)495	6)372	7)532	8)744	9)486	6)276	7)602
6)159	7)399	8)392	9)513	6)168	7)266	8)208
7)434	8)216	9)315	6)348	7)322	8)624	9)882
8)232	9)855	6)126	7)427	8)424	9)252	6)114

PLATE 2

considerably greater. In the cancellation "F-X-P" the experimental group made a greater rate of progress than the control group.

In the psychological tests, the classes "broke even" on the cancellation tests, and in the division and substitution the initial superiority of the control group is sustained throughout, but the progress made by the control group is not greater than that made by the experimental group.

NXCHXJPWNTYFMZYVCZTXHNMJQYTWVCFNHZJMQPNVHJ

NJHVFCPMYHZNQWVFCJNTHXZFCXJNQFWMXNJZHYTWQM

CMQTVPYZWTNPQJCVTYPMJYWXZVCQYTPMHZFJYWVQMC

HMYTWNVJCHZYTPQHNXZXWVPCPHNJCWFZNJXQCJNMFV

MHPWZVXQMHJZYTVCHJFMHTJYFCWQZVXNHJMVYPTWQZ

QTWPCFZVHWVTYXNXJFNHYTWQPZCVNMFJYTMYNTJQHW

MQCVTPMFQFCNHMZCVQYTWHMVZQPMJYTHNPVCQPZWXW

WYZXCJNHMQWYTZHXMCVYXWJFHZPQWYTFJHNFMCVHPZ

NJYTMXQJWNZFHCXVFNHQYTJMVFCXQWTZCVMJNWQYWQ

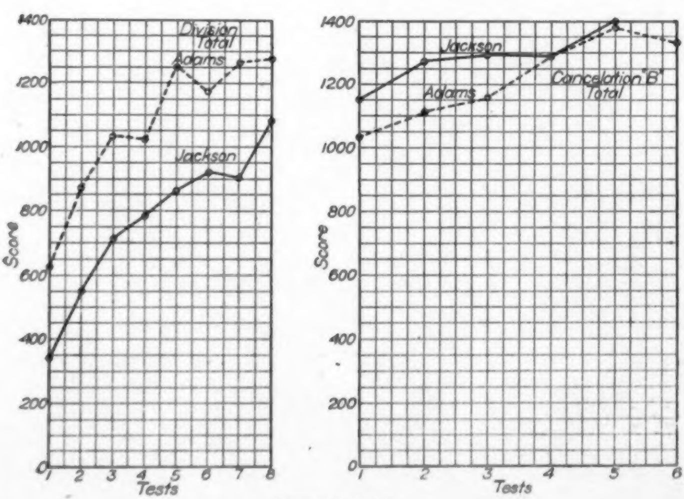
TYZPTPNFHXJMCFVTYWQHZJPMQYFNZYQTHJCVMCYTXC

WTJMNCFVPZJWQTXYZNMVFCJQWYNVCWMQNZJXYCVTPZ

MFNXQXTVWZHCQNTVZFWPHVZHWMYZCHJMYTXNQPWVCZ

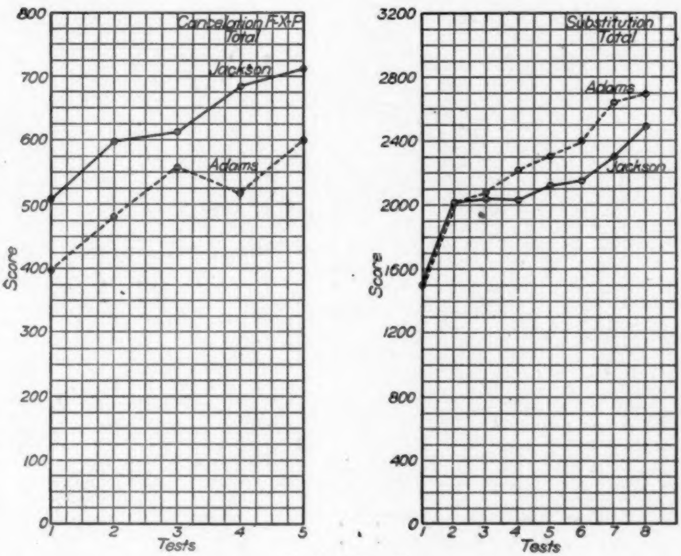
Plate 3.

The pupils in neither room were greatly disturbed by the tests, but the teachers did seem restive. Under the conditions in the experimental schoolroom it was not possible to remove the odors from the air at all times, and the odor at times was objected to by



Plates 4 and 5.

persons entering the room although they would quickly become accustomed to it and feel comfortable. The teacher in this room said that she felt comfortable but that other teachers coming into the room noticed the odor in the air and it annoyed her to be reminded that she was teaching in a room where so-called "fresh air" was not being supplied. The carbon dioxide determinations indicated that in spite of the recirculation of the air a large amount of



Plates 6 and 7.

leakage occurred. The character of the pupils themselves was in some instances such that the odors might naturally be expected to be difficult of removal.

These experiments were, as has already been said, conducted during a period of four months, but with all of the close observation that was given it is impossible to demonstrate physical or mental deterioration due to the use of recirculated air. Neither is it possible to ascribe any discomfort on the part of pupils nor the teacher to this recirculated air. The air washing was not in itself sufficient to remove all odors so that they could not be noticed by persons at the time of entering the room, but it was sufficient to remove them to such an extent that they were not offensive to persons occupying the room continuously. It would seem that if

some method of removing the slight odor could have been utilized that recirculation of air would have been acceptable from every standpoint.

Still further experiments along this same line might be undertaken to great advantage, provided the control of conditions and the character of the observations would be more perfect than in the experiments described, but enough has been done here to show that any deleterious effect of recirculated air upon the occupants of a room during a four months period is extremely difficult of detection, if, indeed, it exists at all.

Synopsis

Experiments to determine if recirculating air was detrimental to the pupils in a room in Jackson school by comparison with a similar room in Adams school which was supplied with a regular ventilating system.

AVERAGE CONDITIONS FOUND

	Temp.	Humidity	CO_2	Dust Count
Jackson	65.3°	46.3%	12.5	105000
Adams	67.2°	42.2%	9.1	225000

Various psychological tests were made to detect differences.

No attempt was made to determine the amount of air leakage but it was admitted as considerable for they could not always keep the windows closed.

Concluded that further tests on similar lines might be advantageous, but this test shows that it is difficult to detect any injurious effects from recirculating air in school rooms.

DISCUSSION

Mr. Carrier: A paper of this character purporting to be a scientific study on the value of ventilation by recirculation should not be given credence by the Society without considerable discussion and criticism. I have checked some of the results alleged to have been obtained. I will recite a few figures to show the physical impossibility of any such results as Prof. Bass attempts to show.

I have no doubt but what Prof. Bass had the best intentions in the world, but he should be careful in accepting his data without sufficient analysis and scrutiny.

In the first place I would call your attention to the CO_2 determination and the distribution of air of Jackson school of 8.9 cu. ft. per

pupil per minute. In the Adams school it was 35.4 cu. ft. per pupil per minute. On the other hand, the air was well distributed in the Jackson school being delivered to each pupil, so that all the air was made to absorb carbon dioxide at its source. Taking the Adams school, with more air supply, there would apparently be an addition of CO_2 of 4 or 5 parts in 10,000. If you reduce the amount of air supplied per pupil and have it supplied from no other source as the paper assumes, it is merely a matter of arithmetic to show that 8.9 cu. ft. would have at least 10 grains addition in CO_2 in 10,000 and probably it would be higher, which would show that there was considerable air coming into the room.

Now apparently there was a great influx of outside air in this room or there must have been some disconnection in the recirculation system that interfered with the apparatus.

There is still another check that can be made on this experiment. Each child gives off approximately from twelve to fifteen grains of moisture. With 8.9 cu. ft. of air we would have probably 1 2/3 grains addition to each recirculation of the air. Where would the moisture content rise to after a few circulations?

Mr. F. K. Davis: The absolute lack of data in this paper as to the condition of the outside air, dust content, CO_2 or temperature when readings were taken is regrettable. There are furthermore, no relative data regarding these rooms, whether the two rooms under test, the controlled room and the experimental room were similar. There is nothing in the paper explaining the differences if any.

The temperature records were taken by thermometers seven feet from the floor. In my opinion this was an error. This would be 3½ feet above the children's heads or above the breathing line. Temperatures at the breathing line and at the floor are liable to vary considerably. I would also like to know if there was any attempt made to get the previous history of the children.

President Lewis: Leakage through a building of ordinary construction amounts to about 15 cu. ft. of air per minute per pupil. A room on the windy side of a building of ordinary construction will take in at least 15 cu. ft. per pupil per minute. Test of recirculation to be conclusive should be made in an air tight room. One other consideration, the question of odors. Granted that after we have been confined for some time in a room in which the air is recirculated we do not object to the odors having become accustomed to them. However, imagine a 24 room school building using such a recirculation system. Each room then will (unless each has a separate fan, washer and duct system) receive samples of the

breathed air and body odors from the other 23 rooms, which they can not get accustomed to and which will be objectionable.

Prof. J. W. Shepherd: Table 1, indicates a lower temperature and higher relative humidity than is customary in most school buildings. This is a desirable condition, and no doubt the lower temperature is possible on account of the 42-46% relative humidity.

Everybody admits that CO_2 itself is not harmful, and since the record for the experimental room has no positive significance as to the oxygen depletion in the air, the record is of no value other than to indicate perhaps something of the efficiency of the air washer.

The dust count is as one would expect and has no significance as to the comparative value of recirculation.

2. The only safe comment one can make on table 2 is that it may possibly indicate a "tendency." Such tables need much cumulative evidence.

It is barely possible that the increase in chest measurement in the experimental room over the control room was due to an effort on the part of the chest to provide the body with more oxygen.

The height and weight increase are both favorable to the control room. This is what should be expected if the percentage of oxygen in the experimental room is enough to keep the children from breaking in health, but not enough for maximum growth.

The paper does not make clear the cause of absences from the two schools.

Too much significance cannot be attached to table 2, because the children in these rooms are about the age when children take on a spurt in growing.

3. Details of the psychological tests are too meager to be commented on.

May I make a few suggestions on the experimental work on recirculation?

1. One hypothesis in such experiments is that the body can do its work quite as well without the usual percentage of oxygen in the air we breathe, as with it. Therefore, in all tests on the comparative value of recirculation, determinations of the percentage of oxygen in the air which is recirculated *must* be made.

2. Again, we must plan for a much longer unit of time for our tests than five or six hours out of twenty-four, and then for but five days out of seven.

3. Table 2 together with a test made by the New York Commission on Ventilation seem to indicate that without serious results

possibly the human body can exist for brief periods in an atmosphere that is reduced in its usual oxygen content. Both sets of experiments point in the direction of probable retarded growth under these conditions.

4. Perhaps children may live for brief periods in a recirculated atmosphere and remain in health, but not grow as rapidly as if they were in an atmosphere of out-door air. Here is the real point to recirculation,—do children thrive as well in recirculated air as in out-door air? Plants need gases from the air in order to grow and yet they will live and remain in good health for several weeks if housed within a glass case that is practically air tight. But they will not grow. The Japanese grow stunted plants, for which they are noted, by carefully restricting their air supply during the period in which they should grow. When the period of growth is past they are forever dwarfed. Perhaps what is true of plants in respect to air during the period of growth is equally true of human beings.

Arthur K. Ohmes: Upon receiving the advance copy of the paper of Mr. Bass on the Recirculation of Air in a School Room, in Minneapolis I analyzed it, with some very queer results. There is, of course, no intention on my part to question the ultimate results of the psychological tests, nor the gain in measurement of the pupils, the attendance, the home conditions, etc. That which interests me most as an engineer are the comparisons of the results of table 1 on the average conditions of the rooms in both schools. Before stating my results I wish to say that the comparisons are based upon the interpretation of "continuous circulation" of the room air which is evidently meant by "recirculation."

As applied in this case it would mean that the air in the room is continuously circulated, or kept in motion by means of a fan, without adding any fresh air to it whatsoever. This continuous air circulation in a room is ordinarily secured by means of desk fans. Desk fans circulate the air in a room at a very much less expense than is possible by a duct system with fans, tempering coils, air washers, etc. To circulate, for instance, 500 cu. ft. of air per minute at 70 degrees temperature at ordinary humidity and normal barometric pressure at say 12 ft. initial velocity, takes but .0053 H.P., but to circulate the same amount of air in a large ventilating apparatus, working with probably ¾ inch static pressure, would take .118 H.P. In other words, 22 times as much.

We are all aware and know the cause of the comfort creating feeling of this circulating air on a hot summer day, and we are equally well aware of the disagreeableness of the air circulation on

a very cold day. On the other hand, it is only possible by means of a complete ventilating apparatus to properly temper, cool, filter and clean the circulating air, and an attempt to show what could be accomplished by means of such an apparatus would prove of interest. Such a test can obviously be made in two different kinds of rooms, one room requiring no artificial ventilation and another requiring artificial ventilation. If a test is made on a room not requiring any artificial ventilation it is obvious that no different results need be expected with a ventilating apparatus bringing into the room fresh air, recirculated air or if the air in the room is simply stirred up by desk fans or other apparatus. In justice to Mr. Bass' paper, however, it should be said that he has taken a room which does require a supply of fresh air. That some supply of fresh air has been secured in this room either through leaky or open windows or through leaks in the circulating apparatus, can be proven by his own figures.

AS TO THE TEMPERATURES MAINTAINED

The experimental schoolroom's average temperature is given at 65.5 degrees with the outgoing air at 70.4 degrees and the incoming air at 56 degrees. There was circulated per pupil 8.9 cu. ft. of air per minute = 534 cu. ft. per hour, which was heated up in the room from 56 degrees to 70.4 degrees on the average. The weight per cu. ft. being very nearly, .078 lbs., this amount of air could absorb per hour .534 x (70.4 — 56) x .078 x .2375 = 142 B. t. u. A child of nine years will give out about 200 B. t. u. per hour. This means that only about 70% of the heat generated in the room by the pupils alone could be carried away by the temperature rise of the air. The test covered the months of January to April inclusive and it is evident that the heat transmission assisted at most times in the removal of the surplus heat. *As the author states, for short periods in the latter days the windows were opened.* What would have happened to the recirculating system in the still warmer months? Or what would happen in a climate (the far greater part of this country) where the average temperature is considerably above that of Minneapolis?

AS TO THE VAPOR CONTENTS

Judging by the low humidity records, it would seem that an active absorption of humidity by the walls during the school hours took place. The walls must again have been dried out during the hours (19 hours per day) when there was no school session. The circulating air was cooled to 56 degrees by adding humidity to the air

from the air washers. A child of nine years will give out per hour on the average 600 grains of moisture. After a three hour session the schoolroom air should have contained not less than

$3 \times 600 \times 39\frac{1}{2} = 71{,}000$ grains of moisture $=10$ pounds approximately and inasmuch as no dehumidifying of air was done the air washer would still add more moisture. But assuming that the air contained only 46.3% humidity with about 65.3 degrees temperature, which with 8000 cu. ft. contents would only be $8000 \times .463 \times 6.85 = 25{,}400$ grains and certainly more than half of this amount must have been in the room in the morning. The surplus of humidity can only be disposed of then in two ways: first, by the introduction of fresh and dry air, second, by the absorption of moisture by the walls. Inasmuch as Minneapolis has pretty nearly the coldest climate, and practically the dryest climate in the United States, both are easily possible. How it would have been in the ocean and great lake regions, or in the warm and moist climate of the southern states, is another question. In sôme localities, no doubt, an active and visible condensation would be caused by continuously circulating the air through an air washer without dehumidification. Condensation on the walls of a room is one of the most unhealthful of conditions, because it offers sure means for the propagation of bacteria and microbes, with possible illness to the occupants.

CARBON DIOXIDE

The class rooms contained probably the usual 8000 cubic feet contents.

In the experimental school there were 39.5 pupils on the average. If during the night time, due to window leakage, the carbon dioxide dilution was brought down to the same amount as contained ordinarily in the outside air then this room would contain, at the usual 4 parts in 10,000,

$$\frac{4 \times 8000}{10{,}000} = 3.2 \text{ cu. ft. of } CO_2$$

One child at nine years generates per hour about 4 cu. ft. of CO_2 or for 39.5 pupils there would be added per hour $39.5 \times .4 = 15.40$ cu. ft. of CO_2. Consequently, if no fresh air is admitted at the end of the first hour the room air would contain $3.2 + 15.8 = 19$ cu. ft. of CO_2. At the end of two hours $3.2 + 15.8 + 15.8 = 34.8$ cu. ft. of CO_2. At the end of three hours $3.2 + 15.8 + 15.8 + 15.8 = 50.6$ cu. ft. CO_2.

This would mean that the room air would contain, neglecting the extremely small amount absorbed by the air washer, at the end

of one hour $\frac{19 \times 10{,}000}{8000} = 23.8$ parts of CO_2 in 10,000. At the end of two hours $\frac{34.8 \times 10{,}000}{8000} = 43.5$ parts in 10,000. At the end of three hours $\frac{50.6 \times 10{,}000}{8000} = 63.3$ parts in 10,000.

The average of CO_2 gas which the room air contains is given as 12.5 parts in 10,000 (see table 1). Mr. Bass also states that the readings were taken in the middle of the session, say between end of the first and second hour. Theoretically, if no fresh air were added there should have been substantially

$$\frac{23.8 + 43.5}{2} = 33.65 \text{ parts in } 10{,}000.$$

With the fresh air school the following calculation would prevail: As before, at starting school the room air contains 3.2 cu. ft. of CO_2. There were added per hour, as stated, 34.5 cu. ft. of fresh air per minute for each of the 41 pupils. This means per hour $41 \times 34.5 \times 60 = 85{,}000$ cu. ft. per hour. This amount of air contained $\frac{85{,}000 \times 4}{10{,}000} = 34.0$ cu. ft. of CO_2. There were added by pupils $41 \times .4 = 16.4$ cu. ft. of CO_2. Total amount of CO_2 in room air and fresh air ($85{,}000 + 8000 = 93{,}000$ cu. ft.) $3.2 + 34 + 16.4 = 53.8$ cu. ft. at end of first hour. With proper dilution this would mean $\frac{53.8 \times 10{,}000}{93{,}000} = 5.8$ parts in 10,000 at end of first hour. At end of second hour: $3.2 + 34 + 16.4 + 34 + 16.4 = 104.0$ cu. ft. of CO_2 in $93{,}000 + 85{,}000 = 178{,}000$ cu. ft. of air, which would mean $\frac{104 \times 10{,}000}{178{,}000} = 5.85$ parts of CO_2 in 10,000. The average between 1 and 2 hours 5.825 parts in 10,000, The average is given as 9.1 parts in 10,000. I am unable to explain the evident discrepancies between theoretical and actual amounts.

Mr. Bass states in his paper: "The carbon dioxide determinations indicated, in spite of the recirculation of the air, a large amount of leakage occurred." If we assume this leakage to be all "fresh air" it seems interesting to follow up this leakage theoretically. It could not have been less actually, and most likely was much more.

In the middle of the session, say at 10:30 after 1½ hours' use, the CO_2 dilution was on the average 12.5 parts in 10,000. Consequently the class room contained by 10:30 a. m. $\frac{8000 \times 12.5}{10{,}000} = 10$ cu. ft. of CO_2 gas. The room air, however, should have

contained 3.2 + 15.8 + 7.9 = 26.9 cu. ft. of CO_2 if no leakage occurred. During this time, however, not less than 20,000 cu. ft. per hour of absolutely fresh air must have been added in order to effect so low a CO_2 dilution in the room as 12.5 parts as shown in the following calculations:

Fresh air in room at 8000 cu. ft. contents contained.......... 3.2
Fresh air introduced from fresh air source (20,000 cu. ft.) contained .... 8.
CO_2 from pupil in 1½ hours 15.8 + 7.9 =.................23.7
Total 34.9

Total air available for dilution purposes for this time is 28,000 cu. ft. which at 34.9 cu. ft. means $34.9 \div \frac{28{,}000}{10{,}000} = 12.5$ parts of CO_2 per 10,000 parts of air.

20,000 cu. ft. means 2½ changes for 1½ hours or 1.67 changes per hour.

Inasmuch as the recirculating apparatus handled during this time 8.9 x 60 x 39.5 x 1.5 = 317,000 cu. ft., fresh air to the extent of 63% of the recirculated air must have "leaked in" somewhere.

In conclusion I wish to say that I cannot yet agree, in view of the above, with Prof. Bass' conclusion "but enough has been done here to show that any deleterious effect of recirculation of air upon the occupants of a room during a four months' period is extremely difficult of detection, if indeed it exists at all."

It may do for Minneapolis in the winter months with its cold and dry climate, but even this is not conclusive for the entire year, and still less for the entire country. We all know that there exists a less demand for ventilation in the cold months of the year, than in the warmer months, and consequently almost every ventilating apparatus is run less actively during the winter months. Our windows in our living and bedrooms are opened but little in the winter compared with the summer months, because we do not need as much fresh air in winter and do not require it for preserving our health. Mr. Bass' paper may have proven that much, but no more.

Prof. Bass: The results of the analyses of the figures by Mr. Ohmes would indeed be "queer," and the figures recited by Mr. Carrier also would show the physical impossibility of the data given in the paper, if the schoolroom used were air-tight.

But the room was not air-tight. There was as in all ordinary schoolrooms, considerable leakage. Carbon dioxide examinations

were included in the examinations largely for the purpose of determining the amount of leakage.

When it is understood that these experiments were conducted under the usual classroom conditions, as far as possible, and that leakage was bound to occur, Mr. Ohmes' analysis shows consistent results, and Mr. Carrier's contention that the figures represent conditions physically impossible disappears.

The discussion by Mr. Ohmes, particularly in the paragraphs under "Carbon Dioxide," brings out by detailed calculation the same point that Mr. Lewis states when he says "leakage through a building of ordinary construction amounts to about 15 cubic feet of air per pupil per minute." Mr. Ohmes' figures indicate that the leakage in the experiment under discussion approximately 8.8 cubic feet per pupil per minute. This is not unusual.

Mr. Carrier makes the statement: "If you reduce the amount of air supplied per pupil and have it supplied from no other source as the paper assumes. . . ." The paper makes no such assumption, and why such an experiment as is described in this paper should be interpreted with such an assumption when we know that an ordinary schoolroom is not air-tight, the author does not understand.

The author cannot agree with Mr. Ohmes' statement that moisture in the form of condensation on the walls is unhealthful because it propagates bacteria. Moisture is incapable of doing harm in this way unless it is first infected and then leaves the walls and gets into the mouths of human beings, carrying with it the contained germs. Such conditions obtain on the walls of a common drinking cup, but scarcely on the walls of a room.

The high carbon dioxide figures at the control school, the Adams, may be explained by the unequal distribution of the fresh air in the room. The samples of air were taken in front of class about 10 feet from the outlet.

Prof. Shepherd's statement that the "CO_2 record is of no value other than to indicate perhaps something of the efficiency of the air washer" is somewhat misleading. In the first place, the absorption of CO_2 by the air washer would not be a proper criterion by which to judge its performance. In the second place, the CO_2 test, as Mr. Ohmes brings out, gives quantitative value to the leakage.

In regard to Prof. Shepherd's comments on the home conditions, the author would agree that merely a "tendency" is indicated. It is also true that table 2, showing gain in measurements, shows nothing significant except the fact that pupils in the two schools showed much the same growth; there is nothing decisive about it.

As Prof. Shepherd suggests more extended experiments would give more light on the problems of growth. If, as he suggests, we plan for a longer time than 5 hours out of 24, and for more than 5 days out of 7, we would be experimenting under different conditions, and the results might be very much different than those shown here.

In reply to Mr. Davis' query, I would say that the two rooms were similar in size and shape. The experimental room had a north exposure and the control room an east and south exposure. The ratio of glass area to floor area was practically the same in both cases. The average outside temperature was 30° F. The CO_2 was taken outside twice and naturally would not vary greatly, 4.5 parts per 10,000. The dust-counts outside were not recorded, as it was not thought average conditions could be obtained without taking more extended observations than was possible. The leakage should be about the same in both rooms.

In conclusion, the author would emphasize the point that the comparison was essentially between two schoolrooms operated under normal conditions. More accurate results could be obtained from two or three human subjects under conditions rigidly controlled, but the results might be difficult to apply to ordinary conditions.

The primary object of recirculating air is to save fuel. The saving of fuel is of little importance in warm climates or in the warmer seasons of cold climates. It is unreasonable to consider recirculation of air for purposes of ordinary ventilation otherwise The conclusion made in the original paper is to be interpreted in this light.

CCCLXVII

TEST OF A CAST-IRON SECTIONAL DOWN-DRAFT BOILER

BY C. A. FULLER

The present condition of the coal market and apparent decrease in the anthracite coal supply, has produced a demand for boilers and appliances which will satisfactorily burn bituminous coal and at the same time pass the rather rigid city requirements in reference to smoke nuisance. This demand has been practically fulfilled for several years on power boilers by means of the Dutch oven type of furnace, which, by reason of its incandescent fire-brick arch, practically consumes all unburned carbon in the gases before the same reaches the comparatively cold tubes or shell of the boiler.

The low-pressure heating boiler manufacturers, both steel and cast iron, have been rapidly falling into line by putting on the market the so-called down-draft smokeless boiler. The various types of these boilers are all constructed along similar lines. In addition to the regular set of grates there is an upper set of water grates upon which the bituminous coal is burned. The hot gases from the burning coal must pass down through the hot bed of coal instead of up through the colder bed as in the regular type and in this passage through the bed of live coals, the unburned carbon is entirely consumed before reaching the parts of the boiler.

This test, the results of which are given below on a Peerless Down-Draft, Cast-Iron Sectional Boiler was made not only to determine its efficiency, rate of evaporation, rating, etc., but also to determine, in a general way, its smoke consuming capacity.

A 4-inch steam main from the boiler was carried through a steam separator constructed of 10-inch pipe and elbows, and then discharged directly to the atmosphere. The water was fed to the boiler through two barrels, both placed above the water line of the boiler. The water was first drawn from the city main to the upper barrel, where it was carefully weighed. It then flowed by gravity to the

second barrel and from there to the boiler. The temperature of this water was also accurately determined.

The firing door of the upper grate was kept open continuously so that the boiler was running at full load capacity throughout the entire test.

The boiler had been running with a slow wood fire for several hours before the test was started and was brought up to full capacity for two hours before readings were begun. The thickness and general condition of the fire was carefully observed at the time of starting. The test was continued for a period of 24 hours and the fire brought back at the close of the test as nearly as possible to the same condition as at the time of starting.

This method is not as accurate as starting with a new fire, but considering the comparatively long period of time over which the test was, the error caused by any difference in the condition of the fire at the starting and closing would be very slight.

In reference to the smoke consuming feature of the boiler, I would say that between the firing periods there was absolutely no smoke visible and it would have been impossible to tell from the appearance of the stack whether or not the boiler was in operation. When fresh coal was fired a very slight evidence of smoke was visible for a period of not more than one minute, so that the operation of the boiler would easily fulfill the requirements of New York City.

The tabulation of the readings and results are as follows:

DATA AND RESULTS OF EVAPORATIVE TEST.

1. Test conducted by Clark, MacMullen & Riley.
2. Kind of boiler, Peerless Down-Draft, Cast-Iron, Sectional.
3. Grate surface, 22 square feet.
4. Water-heating surface, 210 square feet.
5. Date, December 9th and 10th, 1914.
6. Duration, 24 hours.
7. Kind and size of coal, Run of Mine, Bituminous.

AVERAGE PRESSURES, TEMPERATURES, ETC.

8. Steam pressure by gauge, Atmospheric.
9. Temperature of feed-water entering boiler, 52 degrees Fahrenheit.
10. Temperature of escaping gases leaving boiler, 600 degrees Fahrenheit.
11. Force of draft between damper and boiler, .32 inch.

TOTAL QUANTITIES

12. Weight of coal as fired, 3625 pounds.
13. Percentage of moisture in coal, 1.20 per cent.
14. Total weight of dry coal consumed, 3582 pounds.
15. Total ash and refuse, 207 pounds.
16. Percentage of ash and refuse in dry coal, 5.78 per cent.
17. Total weight of water fed to the boiler, 31,350 pounds.
18. Total equivalent evaporation from and at 212 degrees, 36,522 pounds.

HOURLY QUANTITIES AND RATES

19. Dry coal consumed per hour, 150 pounds.
20. Dry coal per square foot of grate surface per hour, 6.82 pounds.
21. Equivalent evaporation per hour from and at 212 degrees, 1522 pounds.
22. Equivalent evaporation per hour from and at 212 degrees per square foot of water-heating surface, 7.25 pounds.

CAPACITY

23. Evaporation per hour from and at 212 degrees, 1522 pounds.
24. Boiler horse-power developed, 44 boiler horse-power or 6100 square feet of heating surface.

ECONOMY RESULTS

25. Water fed per pound of coal fired, 8.65 pounds.
26. Water evaporated per pound of dry coal, 8.75 pounds.
27. Equivalent evaporation from and at 212 degrees per pound of dry coal, 10.2 pounds.

EFFICIENCY

28. Calorific value of 1 pound of dry coal, 14,585 B.t.u.
29. Efficiency of boiler, furnace and grate, 67.9 per cent.

The following is the analysis of the coal used, samples being taken every hour during the test:

	Dry Per cent.	As Received Per cent.
Moisture	.....	1.20
Volatile Matter	22.22	21.96
Fixed Carbon	70.23	69.38
Ash	7.55	7.46
	100.00	100.00
Sulphur (separately determined)	1.14	1.12
British Thermal Units	14,585	14,412

One of the points of interest to be noted is the comparatively high efficiency with the high stack temperature.

I might suggest in closing that the application of an automatic boiler feeder on the feed line would materially assist in a test of this sort as it required considerable attention to maintain a constant water level by hand control.

SYNOPSIS OF PAPER

The author states this test was made not only to determine the efficiency, the rate of evaporation, rating, etc., but also to determine in a general way the smoke consuming capacity of the boiler.

The steam escaped through a four-inch steam main to the atmosphere. The water was fed to the boiler through two barrels, both placed above the water line of the boiler. The water was drawn into the first barrel and carefully weighed; and from this it flowed into the second barrel and from there it was let into the boiler.

Duration of test, 24 hours.

Run of Mine, Bituminous coal used.

Boiler run at atmospheric pressure.

Temperature of flue gases 600 degrees Fahr.

Force of draft between damper and boiler .32 inch.

Amount of coal used, 3,625 pounds.

Amount of refuse and ash, 5.78 per cent.

Total amount of water fed the boiler, 31,350 pounds.

Water evaporated per pound of dry coal 8.75 pounds.

Calorific value of 1 pound of dry coal, 14,585 B.t.u.

DISCUSSION

Mr. A. M. Feldman: I would like to ask if any attempt was made to measure the moisture in, or quality of, the steam during the test?

Mr. C. A. Fuller: Not by any other means than the separator mentioned in the paper. Only twice during the test was there any great amount of moisture removed by the separator and that was when the water level in the boiler was high, due to the uncertainty of hand control, which is not a very good method. It would be better to have automatic control in order to maintain a constant water level.

Mr. Baldwin: The percentage of ash is exceedingly small. I have never known of any quality of coal that had such a small percentage of ash.

Mr. C. A. Fuller: I have no reply to make to that though I do not consider this remarkably low for bituminous coal.

Mr. H. M. Hart: In measuring the grate surface do you count the upper grate plus the lower?

Mr. Fuller: Yes.

Mr. Cassell: I would like to ask if it is possible to use down draft boilers with gas?

Mr. Fuller: Personally I have not had any experience.

Mr. Quay: I would like to ask how the water was returned to the boiler, that is the water that was taken from the separator.

Mr. Fuller: The water that was collected from the separator was put directly back into the second barrel and from there to the boiler. In doing it in this manner it was not necessary to weigh it because the heat in this water was part of the work already done and as the boiler had to evaporate all the water that was weighed, this was put back to the boiler without being weighed to avoid having to deduct it.

Mr. F. C. Bartley: I think there is a doubt about the quantity of heating surface in this boiler. The paper says it contains 210 square feet of heating surface and that it developed forty-four horsepower. That means, a horsepower output of about five feet of heating surface. This is a pretty high rate for any type of boiler. I wonder if there is a mistake in the figures? Also the item "The boiler developed forty-four horusepower, equivalent to 6100 square feet of heating surface." I hardly understand this statement.

Mr. C. A. Fuller: As to the first remark, I would say that over one-half of the surface of this particular boiler and in nearly all cast iron boilers is direct fire surface and for that reason evaporates much more rapidly than in steel boilers, but, as a rule, with a loss in efficiency.

In reference to the second question, 44 horsepower, the equivalent 44 times 34½ gives the total quantity of water. This times 966 gives the total B.t.u.'s contained in the steam. Taking the total B.t.u. and dividing by 250 gives the equivalent radiating surface.

Mr. Davis: If I understand the matter right, it states "Temperature of feed water at entering boiler 52 deg. F." How did they reduce it to 52 degrees?

Mr. Fuller: The temperature of the feed water was taken in the first barrel before the returned condensation was added.

Mr. Davis: The water that went to the first barrel where it was weighed, had a temperature of 52 degrees and from there it went to the second barrel and then to the boiler. Was that all the water that entered the boiler?

Mr. Fuller: I don't quite get your point. If you were to take the temperature of the water in the second barrel after the small

quantity from the separator was added you would find a slight rise in temperature above 52 degrees, but this should not be taken into account, because the boiler has already heated the separator water. The heat that is contained in the water from the separator was added to the water in the second barrel after the quantity and temperature were recorded in the first barrel, that action eliminated all errors.

Secretary Blackmore: Would there be much difference in a test running for 24 hours at different rates of combustion?

Mr. Fuller: We did not attempt to run at any other than the maximum capacity.

Mr. Baldwin: Engineers cannot accept, only in a very general way, the deductions here drawn, because there is nothing to show what the quality of the steam was that was carried out through the escape pipe to the air. Tests should have been made to determine its quality.

Mr. McCann: I would like to ask Mr. Fuller how often the boiler was fired in twenty-four hours, whether it was run on one or two hour periods, and also if anything was done in the way of breaking up of the coal. Was it especially prepared coal or just used as it came in?

Mr. Fuller: In answer to the first question the time of firing was on an average of every half hour. The coal was not specially prepared, but was taken from the regular stock on hand.

Mr. J. S. Otis: I have not had time to analyze this test or figure it out, but what strikes me as very peculiar, is this. If we can get such results as here given, we have obtained something that is very valuable to this Society, for surely such results have never been attained before that I have heard of. One thing strikes me particularly and that is that the temperature of the flue gases are given as 600 degrees. This shows a higher temperature than any boiler manufacturer would concede as being necessary for a boiler in developing its rated horsepower and such a high temperature indicates a very great waste of heat.

Mr. F. K. Chew: I want to approach this test from a different angle. The consumer pays anywhere from $6.00 up for coal, it is not going to be long before he is going to pay more for hard coal, and he will have to resort to soft coal in the suburban districts. I was surprised to see in the Chamber of Commerce Building in Utica a soft coal boiler. The question of the temperature of the flue gases is of secondary importance to the one who buys coal in the country. The fellow who does the buying says he wants the heater that burns a cheap coal. The price of soft coal is reasonable. The day is not so far distant when the soft coal type of boiler will come into general use, if it proves to be economical in operation.

Mr. F. K. Davis: One point I don't see stated anywhere in the paper is the area or size of the smoke chimney.

Mr. Fuller: The chimney was 16 inches square and 55 feet high.

Mr. C. R. Bradbury: A consideration of importance in residences would be the frequency of firing. I would like to know how often a boiler of the type mentioned in the test would need attention to get practical results, for residential heating.

Mr. Lewis: Mr Fuller, can you enlighten Mr. Bradbury as to how often it is necessary to introduce new fuel in a boiler of this type when used for residences?

Mr. Fuller: This would depend somewhat upon the load on the boiler. This boiler was run at full capacity throughout the entire test, and that was the reason for the necessity of firing every half hour.

Mr. Bradbury: I heard some time ago that with such a boiler for ordinary residence work that one would have to be pretty careful about the selection of the size of the boiler, that is, it would need to be of much larger size than would be the case if hard coal is used.

Mr. Bushnell: What would be the result if this boiler was run at such a rate that it was fired only once every eight hours? If the fire temperature falls too low, there is liable to be a waste of unconsumed gases.

Mr. F. K. Davis: Answering Mr. Bradbury's question, there are two types of soft coal in this country. One is non-coking and the other coking coal. To get satisfactory results it would be necessary to use non-coking.

Mr. Bradbury: It is just the opposite.

Mr. Addams: As to the kind of coal required with a down draft boiler, there are about ten thousand kinds of soft coal and you ought to be able to make a boiler to burn any kind and yet consume all the smoke, because coal has to be used wherever it is mined and the boiler has to be adapted to its use. It is up to the engineers to find a boiler that will burn any kind of coal and do it without emitting offensive smoke.

Mr. Lewis: Will you state whether it is practical to run such a boiler on an eight hour period of firing?

Mr. Addams: If your boiler is large enough and you have the right kind of coal that will coke properly, you can run for long periods, because you burn less coal in this type of boiler and with less interference than you do with a fire in which anthracite is used.

The paper giving the particulars of this test has many things of interest. It leads us on in a way that will make us think. There are some things in it which I cannot quite reconcile with ordinary

practice, but which will probably be explained by Mr. Fuller later on.

He states that the boiler has an area of 22 ft. of grate. I infer from this statement that the boiler has 11 ft. in each grate. It is a mistake to consider the measurement of the lower grate which isn't used. This being the case, this comparatively small boiler is burning about 15½ pounds of coal per square foot of grate per hour, a very high rate of combustion. The fact that this boiler was run so hard is evidence that it is entirely too fast for house heating or domestic use. You must have at least four or five inches of fire on the grate and when you add four or five more of coal, you get entirely too deep a bed to draw through with the draft pressure of an ordinary chimney without too frequent attention. I am anxious to know how often it was sliced down?

Mr. Fuller: I could not answer that question.

Mr. Addams: Of course the meat in this whole test is what was the saturation or quality of the steam evaporated from the boiler.

Mr. Fuller: I don't know the saturation off-hand. I will state, however, that in normal running the amount of the drip was very small and about made up for the condensation in the pipe to the separator.

Mr. Addams: A boiler of this type maintains constantly a high temperature and therefore the steam liberating area at the water line needs to be large to avoid pressure convulsions inside the boiler. I have seen a boiler of this type that was run like this for capacity, lift fifty gallons of water in one minute. The question in my mind is, what was the quality of the steam that left this boiler. We ought to know that to be able to place any value on the test.

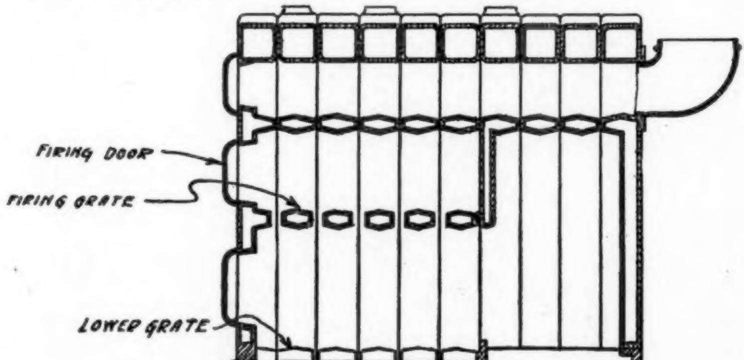
President Lewis: It would seem to be very desirable for Mr. Fuller to have another paper along the same general lines at the Summer meeting covering a further test on this same boiler under house heating conditions.

Mr. G. W. Martin: I have a boiler of this type installed in my own house and I am firing it once every twelve hours in ordinary weather. In extreme weather I stoke it about every eight hours and fill it again in the evening and it gets no attention until the next morning. This boiler is used in a hot water plant, the boiler is about 100 per cent. larger than the necessary size required for the amount of radiation used. I have had considerable experience on down draft boilers and I consider that all soft coal boilers require to be from 75 to 100 per cent. larger than the amount of radiation they are expected to carry.

Mr. Quay: Referring to the test recorded in this paper, I think

there is either something wrong with the test or with the report. This is evident by the number of questions that have been asked. There are four or five points I think should be looked into. One is the square feet of surface compared with the horsepower, the 600 deg. temperature of the gases in the flue also the 67.9% efficiency. He refers to returning the condensation to the boiler, but does not explain how he returned it, nor what the temperature was. Also, his reference to the grate surface and the heating surface is very indefinite. I thought he meant heating surface of the boiler. He should have said heating surface in the radiators if that was what he meant. There are about six points in the paper that are either so indefinite that they are misleading or else they are incorrect. Either the test, or the report of it, in the paper should be corrected, to make it of much value.

Mr. Feldman: I made a test last year on a 250 horsepower Heine Boiler with their expert fireman. With soft coal we obtained per pound of coal 10.4 pounds, evaporation from and at 212 degrees with a flue temperature of 450 to 500 degrees. This test was made with a cheap grade of soft coal and there was no smoke.



SECTION OF BOILER.

Mr. Ellis: I have taken great pains to test different qualities of coal and I find the coal that gives the most gas is known as Penn. gas coal, it gives the highest efficiency. If one puts in a boiler large enough there is no question but what you can run by stoking every 8 or 10 hours. This can be done providing one has the necessary grate surface and lets the fire run very slow. It is better to stoke at least once every eight hours. What we are objecting to in this test is the very high stack temperature. It looks to me as if the results claimed could not have been obtained with such hot flue gases.

CCCLXVIII

EXPERIMENTAL LABORATORY OF THE NEW YORK STATE COMMISSION ON VENTILATION AND A DESCRIPTION OF THE FIRST YEAR'S WORK

D. D. KIMBALL, MEMBER OF THE COMMISSION, PART I

GEORGE T. PALMER, CHIEF OF THE INVESTIGATING STAFF, PART II

PART I

CONSTRUCTION AND MECHANICAL EQUIPMENT OF THE EXPERIMENT ROOMS

The appointment of The New York State Commission on Ventilation was made possible through the generosity of Mrs. Elizabeth Milbank Anderson, who gave to The Association for Improving the Condition of the Poor of the City of New York the sum of \$750,000 for various phases of social investigation, \$50,000 thereof to be expended in an investigation of the problems of ventilation.

The appointment of the Commission was formally announced by Governor Sulzer on June 25, 1913, but an informal notification on May 7th, 1913, had made possible the first informal meeting of the Commission on May 15, 1913, and the first formal meeting was held on June 13, 1913.

The members of the Commission are Prof. C. E. A. Winslow, Prof. Frederic S. Lee, Dr. James Alex. Miller, Prof. Edward Lee Thorndike, Prof. Earle B. Phelps and Mr. D. D. Kimball.

Immediately following the appointment of a Secretary and Chief of Investigating Staff steps were taken to provide laboratory equipment for the conduct of the studies and experiments.

The experimental plant was installed in rooms Nos. 420 and 421 of the fourth floor of the Biological Laboratories of the College of the City of New York, at 139th Street and St. Nicholas Terrace, New York City.

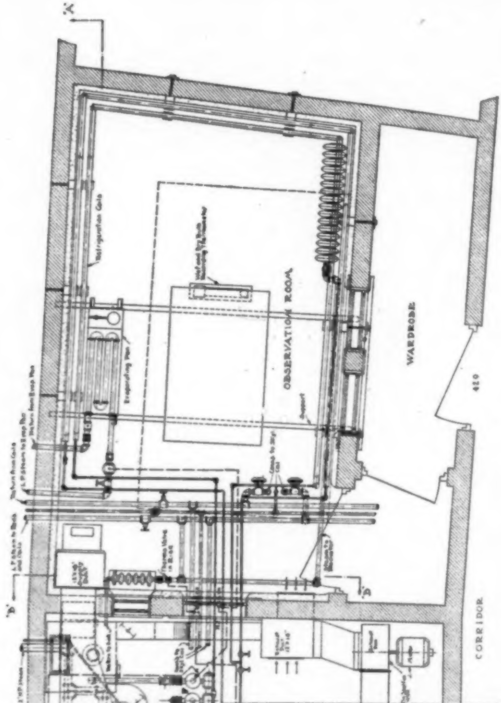


Fig. 1. Plan of Experiment Rooms.

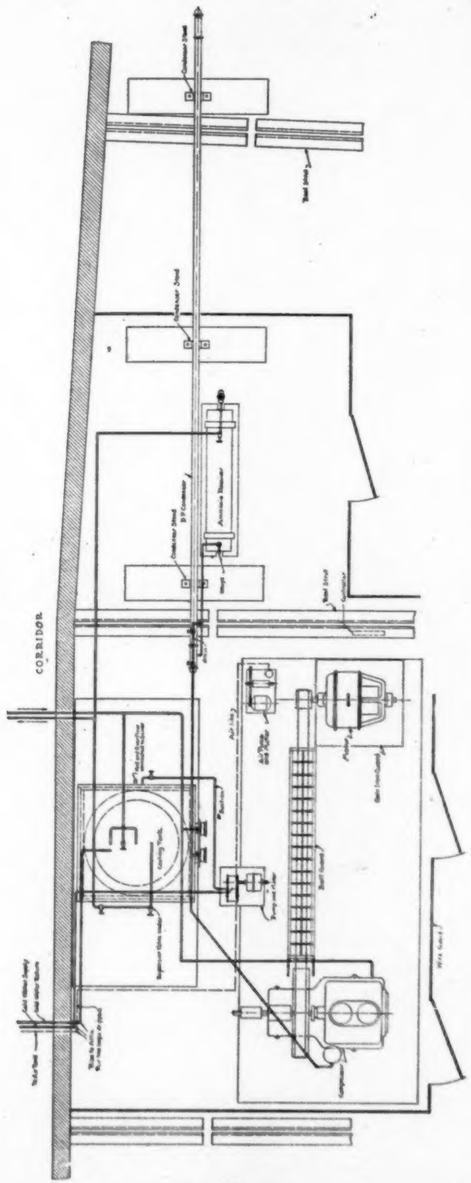
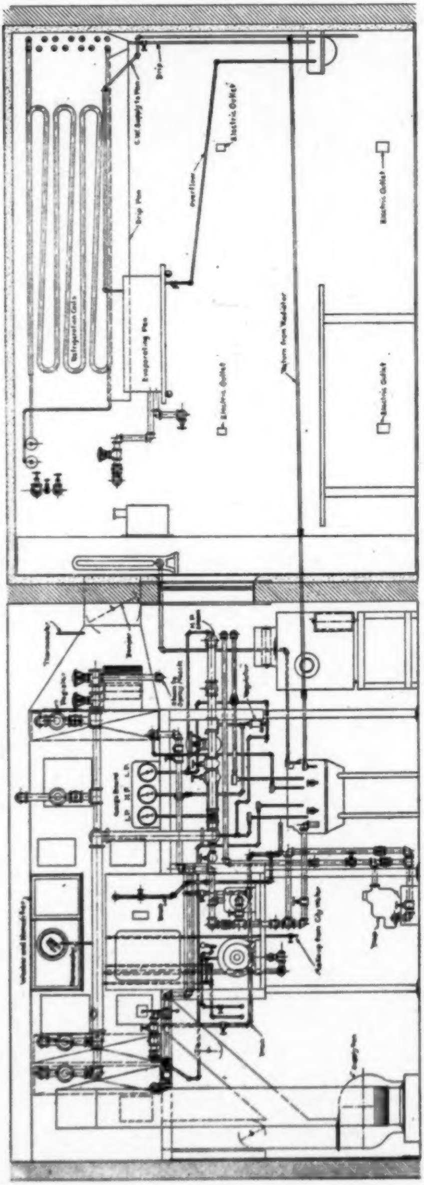
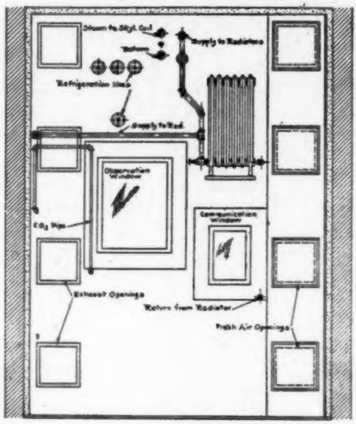


Fig. 2. Plan of Refrigerating Plant.



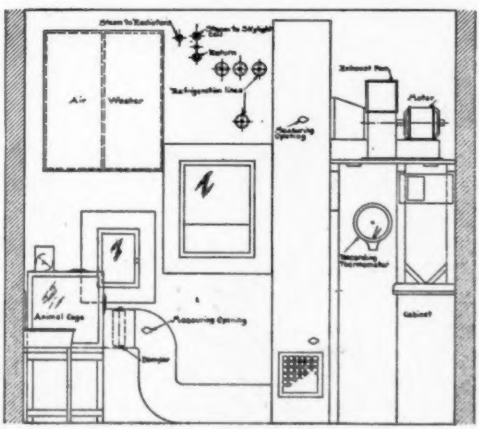
ELEVATION A-A

Fig. 3. Elevation of Experiment Rooms, East Wall.



ELEVATION B-B

Fig. 4. Elevation, North Wall of Observation Room.



SECTION C-C

Fig. 5. Elevation, South Wall of Apparatus Room.

It was aimed to make possible the maintenance of the same or different atmospheric conditions in the two rooms, with temperatures varying from that existing out of doors or less up to 100 degrees F. in zero weather, with humidities varying from the saturation point to practically nothing.

It was later found necessary to modify the extent of the use of the Apparatus Room for experimental purposes because of the space

occupied for apparatus, some of which as, for instance, the dust cages, had not been originally contemplated, and also because of the necessity of the occupation of this room by attendants and operators.

Later a cooling system was added to the plant, the cooling of the Observation Room being accomplished by cooling the water in the air washer and by the use of direct expansion coils on the walls of the Observation Room.

The conditions in the Observation Room are under the control of the observers in the Control Room. They may be manually or automatically maintained.

The air is taken in through a ventilator on the roof and drops to the fresh air fan setting on the floor. The discharge from this fan has two branches, one of which enters the plenum chamber before the tempering coil while the other branch enters beyond the tempering coil. Thus the air may be passed through or by these coils to the air washer or dryer. Over the face of the tempering heater there has been placed a louvred damper so that the heating effect of the heater may be absolutely and immediately eliminated. To the discharge side of the washer the air for both rooms passes through common channels. From this point the apparatus is divided into two parts, one for each room. Each part consists of a chamber in which the air from the dryer and washer may be mixed, from which the air passes through reheaters and thence to the rooms. In the Observation Room this air enters a 12 by 18 inch vertical duct the height of the rooms, having four 12 by 12 inch openings so that the air may enter the room at any one of four levels.

A similar vertical exhaust duct is placed in the Apparatus Room with four openings from the Observation Room and one from the Apparatus Room. From this duct the air is drawn by the exhaust fan, by which it may be discharged into the attic space above the ceiling or back to the fresh air fan for recirculation, if desired. Dampers are provided for regulating the flow and volume of the air. Special openings, 1 inch in diameter, with tight fitting covers are placed at convenient points for air measuring and sampling.

The general drawings shown herewith illustrate the arrangement of the entire plant. Figs. 1, 2, 3, 4, and 5 show the general arrangement of the Experimental Rooms. Fig. 6 is a photographic view of the Observation Room. Detailed descriptions of the various parts of the apparatus follow.

It is not surprising that in the carrying out of a study as complicated as the problem of ventilation many new conditions have been presented which have involved modifications of the original

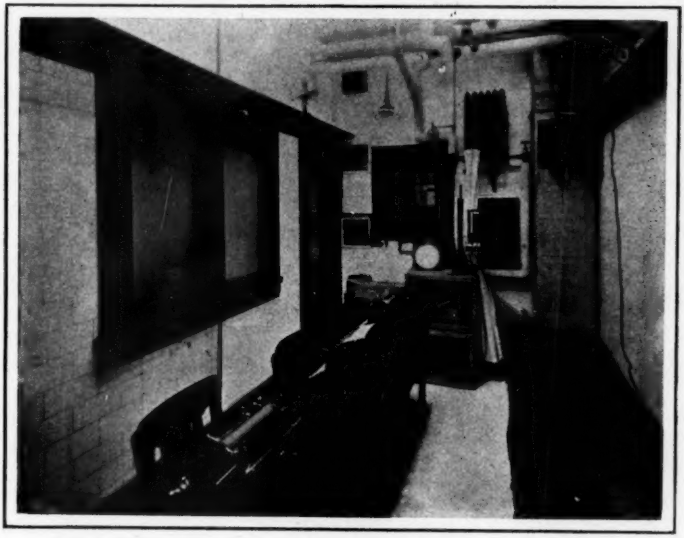


Fig. 6. View in Observation Room, Looking Toward North Wall.

design and additions thereto. The refinement of control desired is exacting and the effect of the occupants and things small in themselves is serious on the temperature and humidity of a room so small.

The fans were furnished by the Massachusetts Fan Co. and are of the Squirrel Cage, or multivane, type, being different only in the direction of discharge, otherwise the same description applies to both fans.

The fresh air supply is a No. 1 top vertical discharge fan with outlet 8 by 9 inches. The exhaust fan is the same size but is a

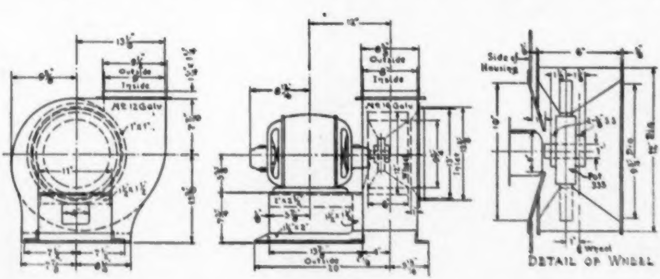


Fig. 7. Details of Supply Fan.

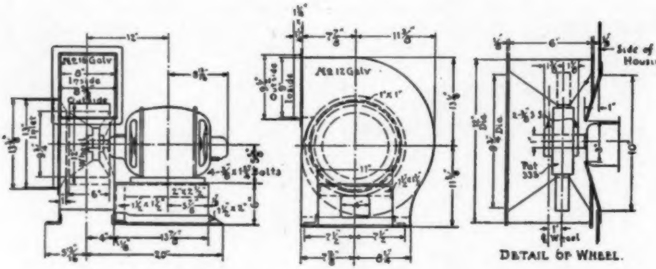


Fig. 8. Details of Exhaust Fan.

top horizontal discharge fan with outlet 8 by 9 inches. These fans are shown in detail in drawings Figures 7 and 8.

The runners are 12 inches in diameter by 6 inches in width, the width being uniform. The fans being of the multivane type, the blades are pitched forward in the direction of rotation, as the action of these fans depends upon the kinetic energy rather than upon the centrifugal force, building up a pressure within the housing. The inlets are large compared to the diameter of the wheel, being 9¾ inches at the upper base of the frustrated cône, while the diameter of the lower base is 13 inches.

The mechanical efficiency of the fans, when operating at 25 per cent. restriction from free discharge, under laboratory tests, is 54 per cent.

The motors, which were manufactured by the Diehl Mfg. Co., are designed to operate these fans at 850 R.P.M., and are equipped

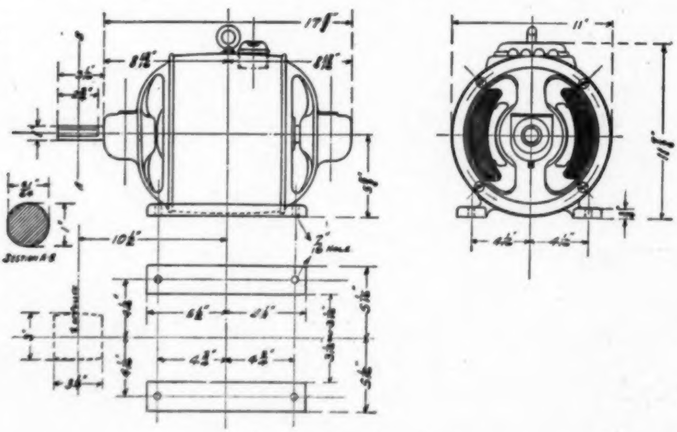


Fig. 9. Details of Fan Motors.

with speed regulators, by means of which the speed may be reduced 50 per cent. through armature resistance.

Each motor is a ½ H. P., 230 volt, shunt wound, Type G, frame No. 302, and is direct connected to the fan. See Fig. 9.

The tempering heater consists of two rows of 40 inch regular vento heater sections, each row six sections wide, sections 5⅜ inches center to center. This heater is common to the work of both rooms.

The two reheaters, there being a separate reheater for each room, each consists of one row of 40 inch regular vento heaters as above, six sections wide, 5⅜ inches center to center. These heaters are illustrated in Figures 10 and 11.

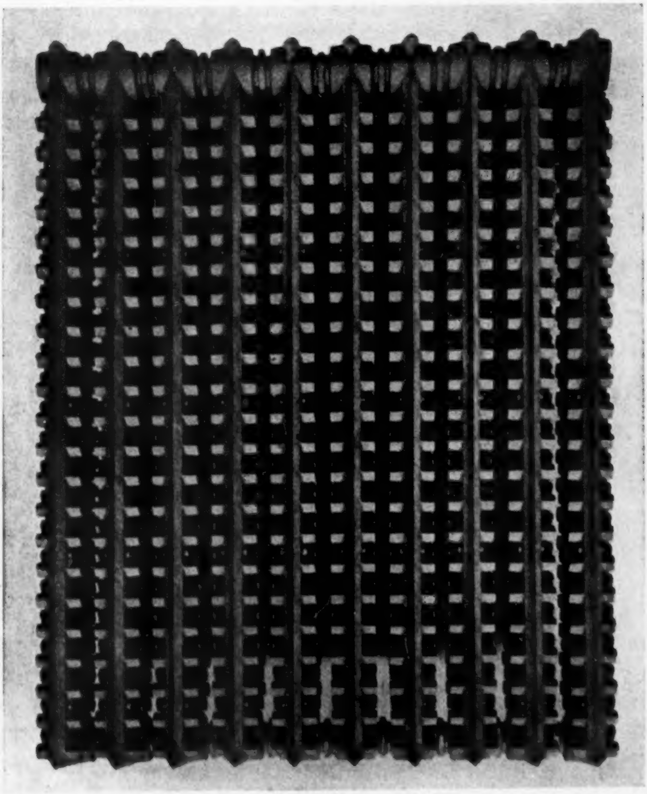


Fig. 10. View of Vento Heating Section.

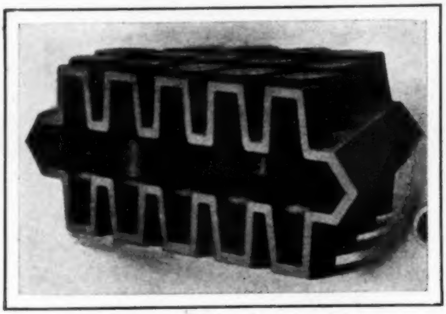


Fig. 11. Detail View of Vento Heater.

The air washer was specially made for this work by the Warren Webster Co., and is 36 inches long, 15 inches high and 36 inches wide, with a water tank 8 inches deep under the washer. This washer is constructed as are most commercial washers, with spray nozzles, eliminators, circulating pump and motor, except that the dimensions are smaller than those of the standard washers.

The circulation pump is of the rotary type with cast iron casings and bronze impeller, driven by direct connected Crocker-Wheeler, direct current 230-volt, shunt wound, electric motor of 1 H. P.

Under the washer is a drying tank to be used for drying the air in the Observation Room when desired. It consists of a galvanized iron box, 36 inches long, 36 inches wide, and 24 inches deep, containing four shelves, each in two sections, covering the entire area of the box, these shelves consisting of ⅛ inch mesh No. 16 galvanized iron wire screen, edges of same bound over ¼ inch galvanized wire, shelves supported on angle and tee irons. It was proposed to dry the air by the use of calcium chloride placed upon the shelves. A 1 inch drain from this tank to sink is provided. The use of this device has not proved practicable with volumes of air used to date.

The above described apparatus is capable of delivering 800 cu. ft. of air per minute maximum with provision for reducing the amount of air to 60 cu. ft. or 30 cu. ft. per minute minimum for each room.

Steam is supplied to the apparatus from a high pressure line brought up from the basement of the Biological Laboratories through a 2 inch pipe, the steam passing through a 2 by 3 inch Standard Regulator Co. pressure reducing valve into a 3 inch pipe with 2 inch supply connection to each section of the heater and a 1 inch connection to the tank of the air washer. One-half inch connections are also made to each of the air discharge ducts for use in humidifying

experiments if desired. From each heater a $1\frac{1}{4}$ inch return pipe is connected into a $1\frac{1}{2}$ inch return main, discharging through an Anderson trap either into the return of the heating system of the building or into the sink.

All steam and return pipes and fittings and the ducts are covered with H. W. Johns-Manville Co.'s 85 per cent. Magnesia sectional covering.

The automatic temperature and humidity controlling system was manufactured and installed by the Standard Regulator Company. This installation originally included a small hydraulic air compressor, which was later replaced with an electric compressor, and the usual air tank, air inlet from intake duct, air piping to thermostats and to valve and damper motors, steam valves and air motor operated dampers. The aim is to provide and maintain automatically any temperature and humidity desired in either room within 1 degree in temperature and 2 per cent. in relative humidity.

The temperature of the incoming air passing through the primary and reheating coils is controlled primarily by thermostat "T-1" located in the intake duct, secondarily by thermostat "T-2" located in the space between the heater and air washer, and finally by thermostats "T-3" and "T-4" located in the Observation and Control Rooms.

Thermostat "T-1" controls diaphragm valve "A" on the outside section of the tempering coil, and is adjusted to open the valve when the incoming air is at or below 35 degrees. It is intended that at no time shall the inner section of the coil or air washer be exposed to freezing temperatures.

Thermostat "T-2" controls diaphragm valve "B" on the inner section of the tempering coil. Its function is to control the temperature of the air entering the air washer and is adjustable to any predetermined degree.

Thermostat "T-3" is located in the Observation Room and controls diaphragm valve "D" on that part of the reheater coil supplying heat to the room. It also controls the diaphragm valves on the direct radiators located in that room. This thermostat has an extended thermostatic member projecting into the room, with the adjustment so arranged that it may be operated from the Control Room.

Thermostat "T-4" is located in the Control Room and operates diaphragm valve "E" in that part of the reheater supplying heat to this room, also the direct radiators located in the room.

The temperature of the space between the inner and outer skylights in each of the rooms is controlled by thermostats "T-5" and "T-6" located in the mentioned space, which thermostats operate

diaphragm valves on the skylight coils. The adjustment of these thermostats can be operated from the floor by means of chain pulls.

Means of creating and controlling humidity is provided in two ways, the first being to condense steam directly in the air washer water, thereby supplying sufficient heat to the water to permit a rapid absorption of moisture by the air passing through the air washer. The heat supplied to the water is controlled by thermostat "T-7," which has its thermostatic member submerged in the settling tank. This thermostat controls diaphragm valve "C" on the steam supply to the water heater. As the temperature of the air entering the washer is held uniform by thermostat "T-2" and the temperature of the washer water is held uniform by thermostat "T-7," the relative humidity of the air leaving the washer is held constant. The relative humidity may be increased by raising the adjustment of thermostat "T-7" or raising the adjustment of "T-2."

A second means of creating humidity was provided for by discharging steam through a nozzle directly into the air after it leaves the reheater coils. The control for this scheme was obtained by the use of hygrostat "H-2" located in the Observation Room and "H-3" located in the Control Room. Hygrostat "H-2" controls diaphragm valve "F" on the steam supply to the nozzle in the duct supplying air to the Observation Room. Hygrostat "H-3" is located in the control room and controls diaphragm valve "G" on the steam supply to the nozzle for the air supply to that room.

The use of the steam nozzles not proving satisfactory an evaporating pan with submerged coil was installed in the Observation Room, the operation of this coil being controlled by hygrostat "H-2."

Hygrostat "H-2" has its thermostatic members projecting into the Observation Room, with the mechanism arranged so that it may be adjusted from the Control Room. See Fig. 12.

Hygrostat "H-2" was intended to be employed to operate lever motor "M-1," and hygrostat "H-3" to operate lever motor "M-2," for the control of the amount of air to be passed through the dryer.

The motive power for the thermostats and hygrostats is pneumatic pressure originally generated by a 2½ by 3 by 6 inch hydraulic air compressor of one cubic foot of free air per minute capacity. This was later replaced by an electrically operated compressor.

The suction for the compressor is taken from outdoors with the purpose of securing air with as low a relative humidity as possible, so as to reduce the condensation in the air lines to a minimum.

The pneumatic pressure is stored at 15 pounds pressure in a 9 by 28 inch galvanized iron tank and is supplied to the thermostats and

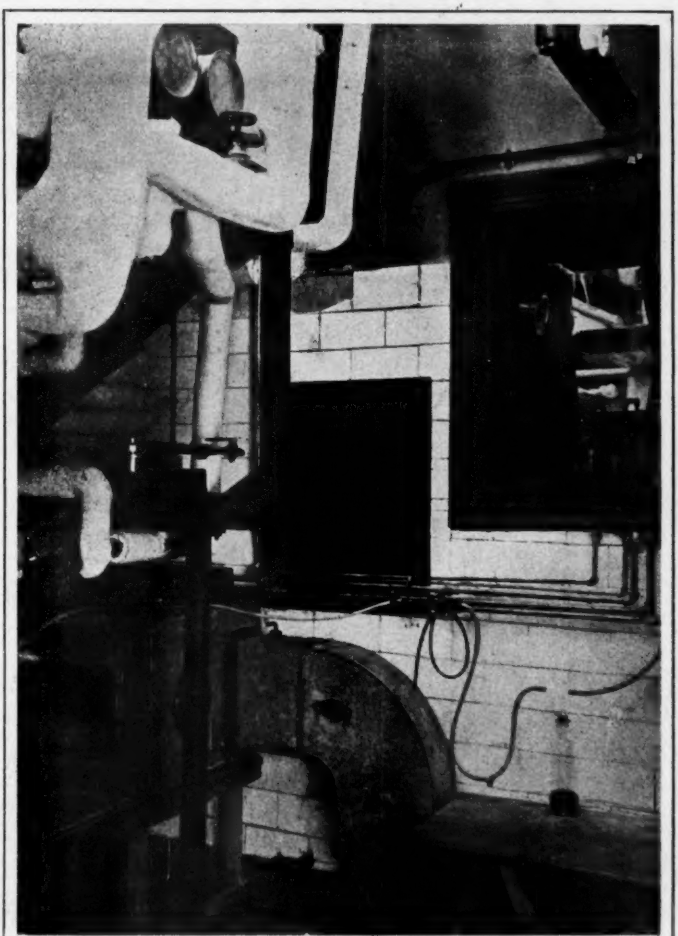


Fig. 12. View of Apparatus Room, Looking Toward South Wall.

hygrostats by means of galvanized iron pipe of small diameter. Each device is provided with a shut-off cock. The control devices are connected to their respective diaphragm valves and dampers by means of ⅛ inch galvanized iron pipe. In the case of hygrostat "H-2" the branch line is connected to both diaphragm valve "F" and lever motor "M-1." Stop cocks are inserted in the line so that either one may be operated independent of the other. Hygrostat "H-3" is connected in a similar manner to valve "G" and lever motor "M-2."

The sheet metal work is constructed throughout of No. 20 gauge galvanized iron except in the case of ducts 12 by 12 inches or smaller, which are built of No. 22 gauge. All joints are made practically air tight. Doors and dampers are felted or close against felted tops and are provided with catch plates to hold same firmly in any desired position.

Measuring openings are provided, having tightly fitting caps which are made air tight by means of thumb screws.

Each fresh air and vent or exhaust opening is provided with an air equalizing register, these registers being equipped with adjustable louvres, key operated.

Three animal cages were constructed, each 36 inches long, 24 inches wide and 16 inches high of No. 20 galvanized iron, all joints being soldered, with removable door in front, full size of front of box, the door being glazed with tight fitting wired glass provided with rubber packing against which the door fits tightly. Later two revolving glass cages with stationary wire cages therein were constructed. These cages are used in animal studies, principally on dust. These boxes were all provided with fresh air and exhaust connections.

The cooling plant, which was later installed, was furnished by the Brunswick Refrigerating Co., and included a single acting compressor of four tons refrigerating capacity. The compressor is of the belt driven, two cylinder, single acting, enclosed self-oiling eccentric drive type, provided with self-acting head pressure relief valve and by-pass between suction and discharge sides. Each cylinder is 4⅞ by 4½ inches.

The condenser for liquefying the ammonia is of the inner tube type, 19 feet long by 6 pipes high and made of 1¼ inch and 2 inch special ammonia pipe.

The liquid ammonia receiver is 48 inches long by 10 inches diameter, with gauge glass, draw-off cocks, inlet and outlet valves, and connections required.

The oil interceptor is 36 inches long and 6 inches diameter.

A full set of 4½ inch gauges are mounted on a cast iron gauge board.

The usual scale traps, valves, cocks, ammonia pipe, fittings, etc., to connect up the above are provided.

The motor is a 7½ H. P., 230 volt, D. C., 650 R.P.M. electric motor of Westinghouse make with rails, sliding base, pulley, starter and 5 inch belt for transmitting the power from the motor to the compressor.

The water cooling tank is 6 feet high by 2 feet 6 inches in diameter. This tank is made of 3/16 inch tank steel. Its insulation consists of granulated cork, water-proof insulating paper, and ⅞ inch boards.

The water is cooled by means of about 200 lineal feet of 1¼ inch continuous electric welded wrought iron direct expansion ammonia pipe.

The water circulating pump has a capacity of 5 gallons per minute and is of the rotary type. An extra pulley is placed on the extended compressor shaft and a 2 inch wide belt is provided for transmitting power from compressor shaft to pump.

The water tank is capable of cooling 75 gallons of water per hour through a range of 40 degrees F.

The cooling of the Observation Room may be accomplished by the use of cooled water in the washer, or directly, as is necessary when stagnant conditions are desired, by the use of the 300 lineal feet of 1¼ inch direct expansion ammonia pipe made up into coils to suit the wall space conditions. These coils are so arranged that, if desired, only one-half the total need be used, but with all the coils in operation, and the compression side working only on these coils a temperature of approximately 50 degrees F. can be produced in

Fig. 13
View of Ammonia Compressor and Rotary Pump for Supplying Cold Water to Air Washer.

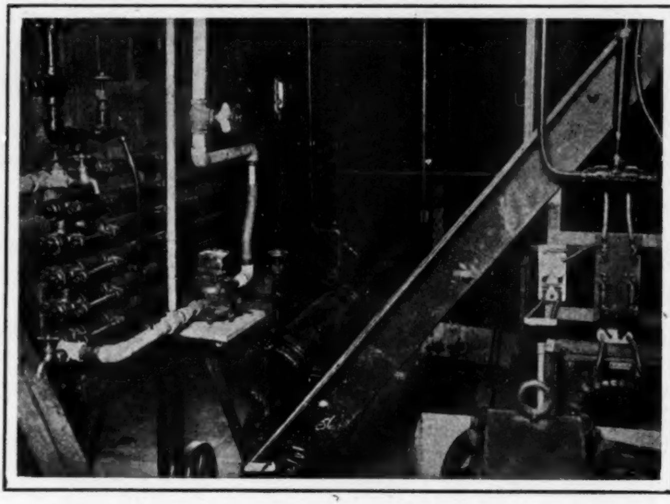


Fig. 14. View of Condensing Coils and Ammonia Receiver.

the Observation Room, when five persons are at work therein, with five forty-watt lamps and a one horse-power motor in operation, and with an outside temperature of close to 90 degrees F.

The ammonia piping is covered with 2 inches of sectional molded cork pipe covering.

The arrangement of this plant is shown in the general drawings and in Figures 13 and 14.

A general idea of the magnitude of this equipment may be gained by the following résumé of the apparatus in regular use.

11 *Fans* with motors attached
- 4 Blowers
- 5 Desk fans
- 1 Fan with psychrometer
- 1 Blower with dust cage

3 *Pumps*
- 2 Water pumps
- 1 Electrical air pump

2 *Motors*
- 1 7½ H. P. for ice plant
- 1 Small motor running dust cages

1 *Ammonia Compressor*

1 *Air washer*—with 18 spray nozzles and float valve water supply controller

Heaters

4 Heating stacks in duct
3 Radiators
2 Ceiling coils

Evaporating pan with steam coil
Steam valves—hand controlled, 32.
Steam traps, 5.
Steam pressure red. valve, 1.
Water valves, 27.
Drain pipe valves, 13.
Ammonia valves, 18.
Electric switches, 8.
Starters, 5.
Electric outlets, 26.
Wall button switches, 6.
Duct dampers, Hand, 19.
Duct dampers, Automatic, 4.
Register openings, 10.
Doors in duct, 13.
Thermometer stations—Mercury, 9.
Thermometer stations—Recording, 2
Air testing stations.
Thermostats, 7.
Humidistats, 2.
Barometer, 1.
Air meter with manometer, 1.
Carbon dioxide machine, 1.
Animal cages, 12.
2 Revolving dust cage containers
5 Stationary cage containers

The Commission is now engaged in installing another experimental plant in Public School No. 51, the Bronx, New York City, which will be described in another paper at a later date.

PART II

EXPERIMENTAL RESULTS OF THE FIRST YEAR'S WORK OF THE COMMISSION

A—OPERATION OF THE VENTILATING PLANT

The construction of the ventilating plant, begun on August 15th, 1913, was sufficiently completed by December 8th so that the rooms could be used for experiments.

The Observation Room at this time had its original tile side walls and plaster ceiling. Heat was supplied by the small wall radiator and moisture was supplied from a ½ inch steam pipe on the open end of which was fixed a Webster air washer nozzle.

The Apparatus Room was equipped to supply outdoor air and re-circulated air at varying temperatures and humidities to the Observation Room.

CHARACTER OF EXPERIMENTS CONDUCTED

In the first series of experiments the efficiency in mental work of four subjects, young men about 18 years old, students of the College of the City of New York, was to be compared in five different atmospheric environments, as follows:

68 deg. Fahr., 50 per cent. relative humidity, ample air supply (about 45 cubic feet per minute per person).

68 deg. Fahr., 50 per cent. relative humidity, no air supply (a stagnant condition).

86 deg. Fahr., 80 per cent. relative humidity, ample air supply (about 45 cubic feet per minute per person).

86 deg. Fahr., 80 per cent. relative humidity, no air supply (a stagnant condition).

86 deg. Fahr., 80 per cent relative humidity, no air supply (a stagnant condition, but with the presence of small electric fans blowing air on the faces of the subjects).

This experiment was thus planned to give information on the subjects' efficiency in (1), a hot moist room as compared with a

cool room; (2), a room with ample supply of fresh outdoor air as compared with a room in which no air at all was supplied; and (3), a hot moist room where relief was afforded by the moving air from electric fans.

Five weeks were to be devoted to this study, the room conditions being changed each day and five separate squads of four men were used, a different squad being observed each week. Of course a natural improvement in their work from day to day due to practice alone was to be expected and to eliminate this practice effect in computing the final averages the sequence of room conditions was varied each week. Thus Room Condition A would occur with the first squad on Monday, with the second squad on Tuesday, with the third squad on Wednesday, with the fourth squad on Thursday and with the Fifth squad on Friday.

The men were kept in the Observation Room from 2.30 to 6 P. M.

OPERATION OF VENTILATING EQUIPMENT DURING FIRST TWO WEEKS

The 68-degree condition with air supply was obtained fairly close on the first day. The outdoor temperature was 46 degrees. With the aid of the heaters and washer this temperature was raised to about 65 degrees at the entrance to the Observation Room. The dry and wet bulbs within the room both gradually rose during the afternoon to 71 degrees and even though the steam supply to both heater and washer was cut off later on in the afternoon yet the temperature of the room could not be lowered.

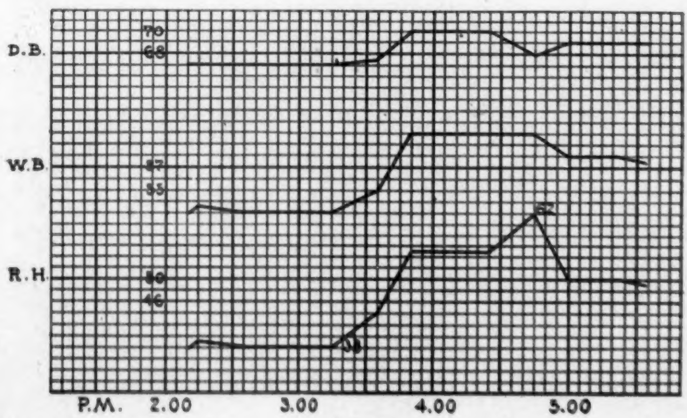


Fig. 1

Effect on Dry Bulb Temperature of Room caused by Raising Temperature of Wash Water to Increase the Humidity. (12/19/13.)

In the second week the outdoor temperature on the day that the above condition was again used was 37 degrees. Air was forced into the Observation Room at the full speed of the fans from 1.30 P. M. till 2.30 and the temperature was reduced from about 75 degrees, the common temperature of the rooms on this floor of the building, to 67 degrees. With the temperature of the washer water at 67 degrees it was difficult to raise the relative humidity above 35 per cent. When heat was turned on the washer at 3.15 the wet bulb was raised but the dry bulb also went up. (See Figure 1.) Once the temperature had risen above the desired point it could not be lowered quickly owing to the lagging heat given off from the heating stacks.

The 68-degree stagnant condition was tried during both weeks by cooling the room down with outside air until the subjects entered the room and then cutting off all air supply. On December 12th the temperature at 2 P. M. had been lowered to 71 degrees. This gradually rose during the four hours experiment up to 72.5 degrees. The wet bulb showed no change throughout this period. On December 18th the walls of the room evidently had not been chilled as thoroughly as on the 12th for the dry bulb rose from 71 degrees at the time the four subjects and the observer entered to 76 degrees at six o'clock. A corresponding increase in the wet bulb raised the relative humidity from 31 to 40 per cent.

To attain the 86-degree condition with 80 per cent. relative humidity and air supply heat was turned on the ceiling coils at 10 A. M. Owing to the overheating of some of the apparatus the blowers

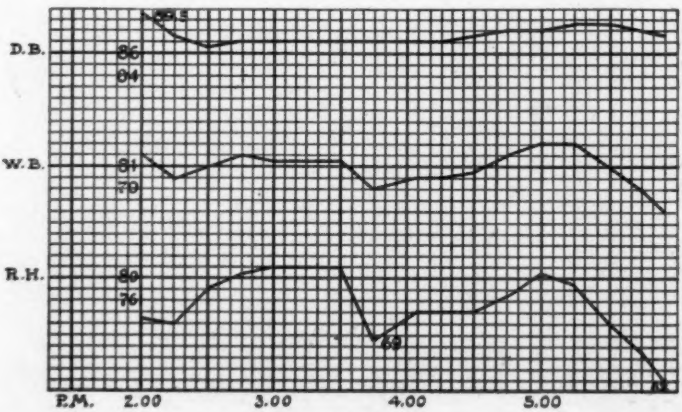


Fig. 2
Showing Difficulty in Raising Wet Bulb Temperature without Raising Dry Bulb. (12/17/13.)

could not be used till after 2 P. M. and consequently the desired conditions were not attained till 3.30 P. M. On December 17th the blowers were started about 1 P. M. and in an effort to raise the temperature quickly the room was slightly overheated at 2.30. The difficulty encountered in keeping the room at 86 degrees with 80 per cent. relative humidity is well illustrated by Figure 2. At 2.45 P. M. the temperature of the wash-water was reduced in the hope of lowering both dry and wet bulbs. By 3.45 the wet bulb had dropped to 79 degrees but the dry bulb temperature remained unchanged. The temperature of the washer water was then raised slightly to restore the humidity and in consequence the dry bulb rose from 87 degrees at 4.15 to 88.5 degrees at 5.15. The lowering of the wash-water temperature at 5.15 again caused a drop in the wet bulb but did not change the dry bulb materially.

The attempt to carry out the hot, moist condition without air supply proved wholly unsuccessful as shown in Figure 3. Although

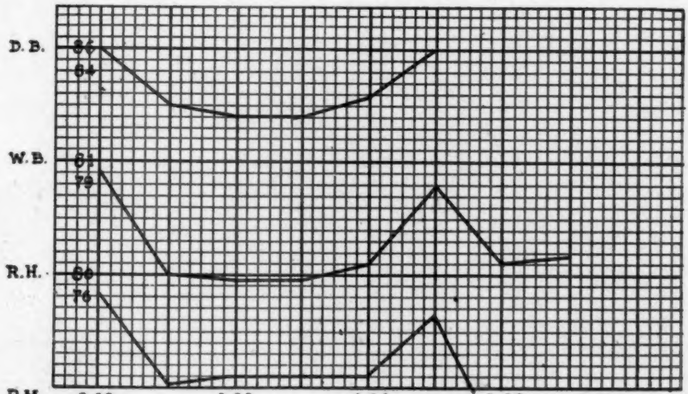


Fig. 3
Showing Necessity of Raising Walls of Room to Desired Temperature in order to Maintain Constant Air Temperatures. (12/9/13.)

the hot, moist air was forced into the room for over an hour yet as soon as the fans were stopped, both wet and dry bulbs fell off immediately. At 4.15 P. M. air was re-circulated until 4.40, restoring the temperature and humidity temporarily. On shutting down the fans again, however, the temperature fell abruptly. The heat from the wall radiator in the room was wholly inadequate to keep up the temperature. The steam nozzle annoyed the subjects with the hissing sound and by the ejection of water on the table tops. Furthermore the sudden addition of the warm, moist air at 4.15 pro-

duced an objectionable accumulation of condensation on the walls and skylight.

On December 15th no attempt was made to produce the hot, moist condition from within the room itself. Re-circulated air through the heaters and washer was used throughout the afternoon.

The hot, moist, stagnant conditions with electric desk fans came on December 10th and 16th. The failure to bring the room to 86 degrees by 2 P. M. was due to the long time required to remove condensation water from the system as steam could not be turned on full until the water had been drawn off. The temperature was raised to 88 degrees in an unsuccessful attempt to raise the relative humidity. Air was re-circulated on this day and on the 16th. Figure 4 again illustrates the difficulty in maintaining 80 per cent. relative humidity with 86 degrees dry bulb.

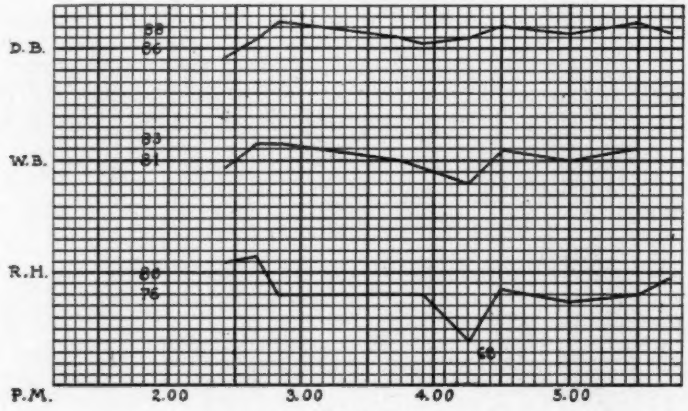


Fig. 4
Showing Difficulty in Raising Wet Bulb Temperature without Increasing Dry Bulb. (12/16/13.)

LIMITATIONS OF PLANT

The following limitations were demonstrated during the two weeks' operation:

1. Inability to maintain high relative humidity without overheating.
2. Inability to cool quickly when overheating did occur.
3. Inability to bring the room to proper conditions without using large volumes of air requiring several hours time.
4. Inability to maintain desired temperature and humidity under stagnant conditions.

ALTERATIONS IN EQUIPMENT

For experimental work of this character a very close approximation to the desired air condition was necessary and it was evident by the end of these first two weeks that the control of the plant was handicapped in several ways. To meet these objections, therefore, and also to improve certain other shortcomings that were apparent, the following alterations were made at this time:

1. Complete insulation of the Observation Room with 2 inches of cork board and ½ inch coating of cement with a smooth white cement finish. (This was done to cut down heat loss through the walls and perhaps makes unnecessary the increase of radiation surface.)

2. Installation of a humidifying pan within the Observation Room. (See Fig. 5.)

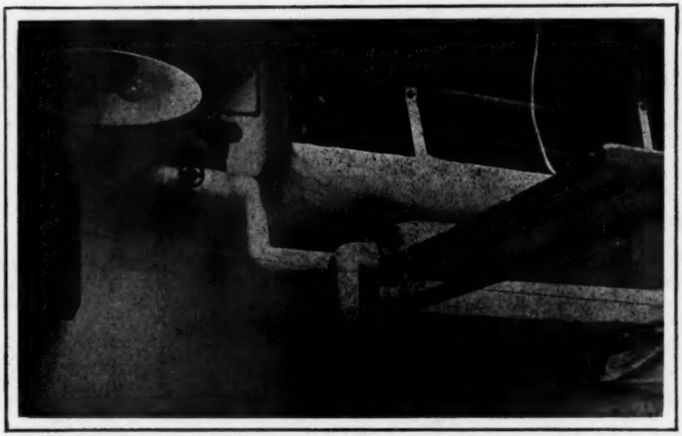


Fig. 5

Humidifying Pan with Steam Connections in the right foreground. Just above this is a section of the Refrigerating Coils with Insulated Drip Pan beneath. At the left is a side view of the Air Meter.

3. Installation of a 4½ ton refrigerating plant with cooling coils in the Observation Room and with a cold water tank connected to the spray nozzles of the air washer. (This would permit direct cooling of the Observation Room and also the cooling of air passing through the washer.)

4. Complete insulation of air ducts, washer, etc., in the Apparatus Room with asbestocel sheets 1½ inches in thickness.

5. Construction of a duct from the supply blower direct to the washer permitting air to be by-passed around the pre-heating stacks.

OPERATION FOLLOWING THE INTRODUCTION OF ADDITIONAL EQUIPMENT

The insulation work and changes of a minor character were sufficiently completed by the latter part of January so that the experiments could be resumed. The benefits of the changes introduced were not fully realized during January and February owing to the injection of other difficulties, principally the failure of the air system controlling the thermostatic valves. The water pressure in the building fluctuated so that at times not over 10 pounds compression could be procured from the hydraulic pump. An improvement was effected here by installing an electric air compressor with an automatic cut-out.

The refrigerating plant was first used on February 26th and its effect is illustrated by the temperature ranges on the cool days with re-circulated air before and following its use. (See Table 1.)

TABLE 1

Maintenance of 68 degrees temperature with recirculated air in the Observation Room before and after installation of the Refrigerating Plant

Date		Temperature, Degrees Fahr. Minimum	Maximum	Average	Air Movement Cubic feet per minute per person
Dec.	12	71	72.5	71.5	0
	18	71	75.5	74	0
Feb.	3	70	75.	72.5	45
	11	65	71.5	69	45
	20	72.5	74	74	20
		Refrigeration used from Feb. 26th.			
Feb.	26	65	68.5	67.5	40
Mar.	2	66	68.5	67	40
	10	64	70	68	40
	19	68.5	72	70	40
	24	65.5	71	69	35

SOUND INSULATION OF AMMONIA COMPRESSOR AND MOTOR

One cause of delay in the use of the refrigerating plant was the difficulty encountered in preventing the noise of the compressor and motor from reaching the classrooms on the floors below. The motor and compressor were first set on a platform 5 feet by 12 feet 6 inches by 7½ inches thick. The construction beginning at the floor included a ½ inch layer of cement, 2 inches of cork board and a framework of 4 by 6 inch lumber filled with concrete. The noise of running was plainly evident on the 1st, 2nd and 3rd floors of the building. The platform had to be re-built entirely and the arrangement which finally proved successful in eliminating the transmission of noise to the floors below consisted of 1 inch of hair felt next to the cement floor, three 4 by 6 inch sleepers bridging two roof trusses and four floor beams to distribute the

load, a 7/8 inch wood platform, 1 inch of hair felt, a layer of 28 inch gauge galvanized iron, 1 inch of hair felt, 2 inches cork board, the 4 by 6 inch framework with the concrete filler, and a 1 inch hair felt and rubber pad between the metal bases and the framework.

LOW CO_2 CONTENT OF RE-CIRCULATED AIR

Owing to considerable time spent in making satisfactory the construction, location and control of the evaporating pan, humidification from within the Observation Room was not practicable until the middle of May. From this date on it was possible to produce a constant temperature and humidity under stagnant conditions. Thus during January, February, March and April re-circulated, washed air was used as the nearest approach to the stagnant condition.

As measured by the carbon dioxide, however, the re-circulation of the room air through the washer and back to the room again represented outside air much closer than it did used air. The carbon dioxide content on these days never rose above 8.5 parts per 10,000. This fact was largely due to the necessary location of blowers and angles in the ducts, all of which favored air leakage.

In Table 2 is given a summary of the carbon dioxide analyses for the different room conditions during the eleven weeks of the first series.

TABLE 2

Average of Carbon Dioxide Analyses for the 11 Weeks of the First Experiment

	CO_2 parts per 10,000		
	2 P. M.	4 P. M.	6 P. M.
Hot, moist, outdoor air	6.0	5.5	5.5
Hot, moist, recirculated air	6.0	5.5	7.0
Hot, moist, recirculated air, with electric fans	5.0	8.5	8.5
Cool, outdoor air	6.5	..	6.5
Cool, recirculated air	6.5	..	8.0

In view of the above the first experiment reduced itself to a study of three air conditions instead of five, namely:

(1) 68 degrees temperature, 50 per cent. relative humidity, ample air supply.

(2) 86 degrees temperature, 80 per cent. relative humidity, ample air supply.

(3) 86 degrees temperature, 80 per cent. relative humidity, ample air supply, with the additional air movement created by electric desk fans.

OPERATION DURING THE SPRING MONTHS

During April and May it had been arranged to study two variables:

(1) 68 degrees temperature, 50 per cent. relative humidity, ample air supply; and

(2) 86 degrees temperature, 80 per cent. relative humidity, stagnant air.

Owing to the delay in the use of the evaporating pan, this second condition was not attained until May 13th, and re-circulated air was again used up to this date. The creation of the true stagnant condition is indicated by the sudden increase in the CO_2 analysis on this date which will be found in the chronological summary of Room Conditions on page 46.

DEHUMIDIFICATION

In this period the ventilating plant encountered a new problem. So long as the outdoor air was below the temperature desired in the experimental chamber, it was a simple matter to apply heat and moisture. But as soon as the temperature out of doors approached or exceeded that required for the room, it became necessary to cool and dehumidify. The effect of this seasonal change on the method of operation is shown in Table 3. Thus during the first week it was necessary to add 1.55 grains of moisture to each cubic foot of air, during the second week 1.24 grains, during the third week to remove 0.32 grains, and during the last week to remove 1.94

TABLE 3

Effect of Seasonal Change on Plant Operation
(Average of daily readings taken at 4 P. M.)

Period	Out of Doors						Resulting Condition in Observation Room			
	Temperature		Rel. Hum. %	Grains moisture per cu. ft.	Moisture Deficiency	Moisture Excess	Temperature		Rel. Hum. %	Grains moisture per cu. ft.
	Dry Bulb	Wet Bulb					Dry Bulb	Wet Bulb		
Condition desired							68	57	50	3.74
Mar. 30-Apr. 3.	45	39	64	2.19	1.55		68	57	49	3.67
Apr. 20-Apr. 24.	59	45	45	2.50	1.24		69	58	51	3.94
May 4-May 8.	63	53	64	4.06		0.32	68	58	53	3.90
May 25-May 29.	77	65	57	5.68		1.94	68	58	51	3.81
June 8-June 12.	81	68	51	5.75		2.01	67.5	59.5	62	4.56

grains. The first week in June overtaxed the powers of the equipment, as the relative humidity in the room could not be reduced below 62 per cent, the excess moisture in the outdoor air being 2.01 grains.

Dehumidification was accomplished by chilling the spray water in the washer. When necessary the temperature of air passing the washer was restored by a small amount of heat from the reheater. Calcium chloride was also tried in this connection, the dry crystals being placed on trays in the space provided beneath the washer. Its effect in extracting moisture from a stream of air passed over it at the rate of 225 cubic feet per minute was scarcely noticeable. Furthermore,, the handling of the chloride was disagreeable and time consuming.

TEMPERATURE CONTROL

The original installation had been planned with a view to making air conditioning units as automatic as possible. Thermostats and humidostats were located to control the steam supply and are indicated on the general drawings as follows:

Thermostat A (before first heater) controls steam to first heater.

Thermostat B (after first heater) controls steam to second heater.

Thermostat C (in washer) controls steam to washer water.

Thermostat D (in window) controls steam to reheater and Observation Room wall radiator.

Thermostat E (near skylight) controls steam to ceiling coils in Observation Room.

Humidostat (in window) controls steam to steam jet after reheater and jet in Observation Room. This also operates the damper controlling the course of the air through the washer and its by-pass.

This arrangement, though useful in many ways, did not prove successful. The preheater had been designed so that 800 cubic feet of air per minute could be raised from temperatures around the freezing point up to about 90 degrees. This requirement demanded a much larger radiating surface than would be called upon ordinarily. Furthermore, the air volume commonly used in the experiments was only 225 cubic feet a minute, or much lower than the maximum provided for. In cold weather or below 40 degrees, Thermostat B was very dependable in keeping a proper amount of steam on the second heater. When the outdoor temperature rose above this point, however, the steam supply to the heater even at low pressures was sufficient to cause overheating of the air stream

for at each opening of the valve the stack was suddenly flushed with steam-producing excessive heat. This effect was partially, though not entirely, overcome by cutting down the opening to a minimum with the hand valves.

Thermostat A was used only in extremely cold weather. Its location in the cold air stream was designed to protect the first heater against freezing, but with comparatively small volumes of air this protection proved unnecessary. This heater was rarely used as there was no automatic means of cutting off the steam once the air stream had received its necessary quota of heat.

Most reliable of all times has been the thermostat in the washer. No hand control other than the setting of the thermostat dial has been necessary here.

The thermostat and humidostat located in the window between the two rooms have been of no assistance whatever in operating automatically the reheater, and likewise the heater and evaporating pan within the Observation Room. This fault is due probably more to the location of the expansion rods than to the mechanism itself. If the rods could be located in the center of or at a representative point of the room bathed constantly by air currents, then the response would probably be fairly quick and accurate, but even then the completeness of their action causing, as it does, a tight closing or wide opening of the diaphragm valve renders this type of instrument unsuitable for precise experimental work.

For example, in operating the dampers controlling the course of the air through the washer, the wet bulb of the humidostat contracts indicating dryness, and thus closes the by-pass dampers, thereby sending the air stream through the washer. As the humidity in the room is made up the wet bulb expands and the damper in front of the washer is closed and the by-pass opened. This action permits the air stream without any humidification to pass on into the room. The humidity thus drops very quickly and the wet bulb temperature of the room falls several degrees before the wet bulb of the thermostat can sense the change and readjust the dampers.

This method of operation causes a fluctuating wet bulb temperature in the room in place of the desired constant conditions. We have found that a much more delicate adjustment of humidity can be had by altering the temperature of the wash water.

The same objection to the positive action thermostat applies in the case of the evaporating pan. The sudden complete opening of the steam supply valve to the coils in the pan results in excessive evaporation, whereas the complete shutting off of steam allows the water to cool down too low.

It must be borne in mind throughout this discussion that objections raised against apparatus on an experimental scale might cease to be such in large installations. The demands of the experiment chamber are extreme, there being occasions where such factors as moving air, noise and temperature fluctuations are undesirable. In practice all of these limitations would not exist. Of primary importance in this work is the production of the air condition desired. The method of producing the condition is of secondary consideration.

The control of temperature has therefore been obtained in this experimental work by balancing excesses. To produce 68 degrees and 50 per cent. relative humidity under a stagnant condition actually requires but a small amount of cooling to neutralize the heat given off by the subjects and electric lights in the room. In fact the amount of cooling to take care of two to four people is so slight that the refrigerating plant need only be operated intermittently

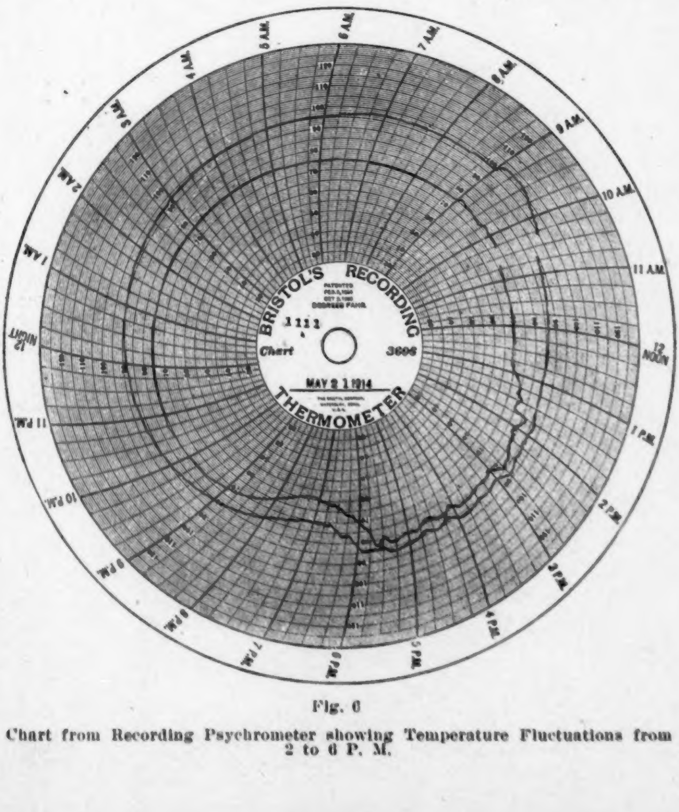


Fig. 6

Chart from Recording Psychrometer showing Temperature Fluctuations from 2 to 6 P. M.

at its lowest output. This intermittent operation, however, causes temperature fluctuations and requires undue attention in starting and stopping the compressors. (See Fig. 6.) It is much simpler to operate the compressor continuously at a low output and neutralize the excess cold with the heaters. This, in short, is the method that has been found most successful in our work.

In producing a stagnant condition the three factors, refrigeration, heat and moisture, are worked against each other by hand control. It has not appeared at all feasible to manage this problem automatically because of the number of independent variables involved. These may be listed as follows:

1. Refrigeration—lowering temperature and removing moisture by condensation.
2. Heat—raising temperature and removing moisture by evaporation.
3. Humidification—producing moisture and raising temperature.

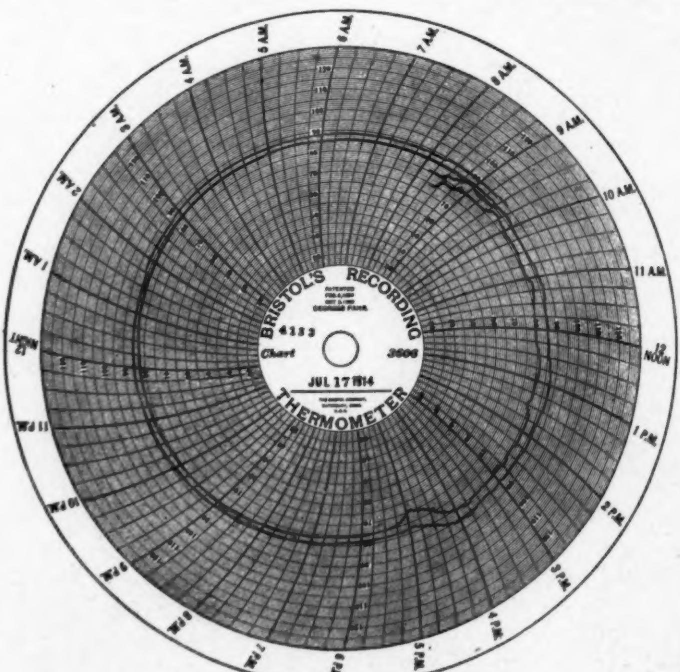


Fig. 7.

Psychrometer Chart showing Absence of Sudden Temperature Fluctuations from 9 A. M. to 4 P. M., as compared with Fig. 6.

To further interfere with any attempt at automatic control, it must be remembered that space in the experiment room is at a premium and thermostats cannot be located at the most representative point and even if so located the thermometer bulbs could not be bathed in air currents when stagnant conditions in the experiment room are scheduled.

When operating with moving air, likewise, the ice coils on the walls of the Observation Room and the heater are frequently called into play to neutralize changes in the air stream and thus insure constancy. (See Fig. 7.)

TEMPERATURE OBSERVATIONS

The nature of the experiments conducted made it unfeasible for an observer to enter the Observation Room at will to take temperature readings with a sling psychrometer. This method was therefore abandoned during the very first week of the experiments in favor of reading thermometers suspended close to the Observation Room window. Various methods of circulating air past the bulbs were tried, all of them introducing some objections, either of noise or of stirring up too great air movement within the room.

Up to February 27th, therefore, the wet bulb determinations were somewhat irregular, as numerous check readings in different parts of the room will indicate. (See Table 4.) From the first of March, all temperature observations were made with an air current passing the bulbs. A small fan on a battery circuit was located in the Apparatus Room, pulling air through a small sheet-metal duct, 3 inches in diameter, which pierced the window between the two rooms. The wet and dry bulb thermometers were suspended just at the opening of this duct. Thus, to take an observation, the small fan was started up inducing a current of air from the Observation Room past the bulbs and through the duct into the Apparatus Room. The thermometers could be read from the window.

TABLE 4

Temperature Observations at Various Points in Observation Room

Date	Time	Location	Dry Bulb Temp.	Wet Bulb Temp.	Rel. Hum. %
12/15/13	5:30 P.M.	Window	87	82.5	82
		Sling psychrometer center of room	86.5	81.5	81
2/3/14	3:45 P.M.	Window	71.5	59.	47
		Wall opposite window	71.5	61.5	57
2/9/14	4:00 P.M.	Window		77.	
		Center of room 7 ft. from floor		79.	
2/10/14	4:00 P.M.	Window		80.	
		Center of room 3 ft. from floor		79.	

TEMPERATURE RECORDS WITH RECORDING PSYCHROMETER

On May 13th, a recording psychrometer of the Bristol Company pattern was installed in the Observation Room. The wet and dry bulbs, with a small fan connected, were attached to the bottom of a platform suspended in the center of the room. The bulbs were

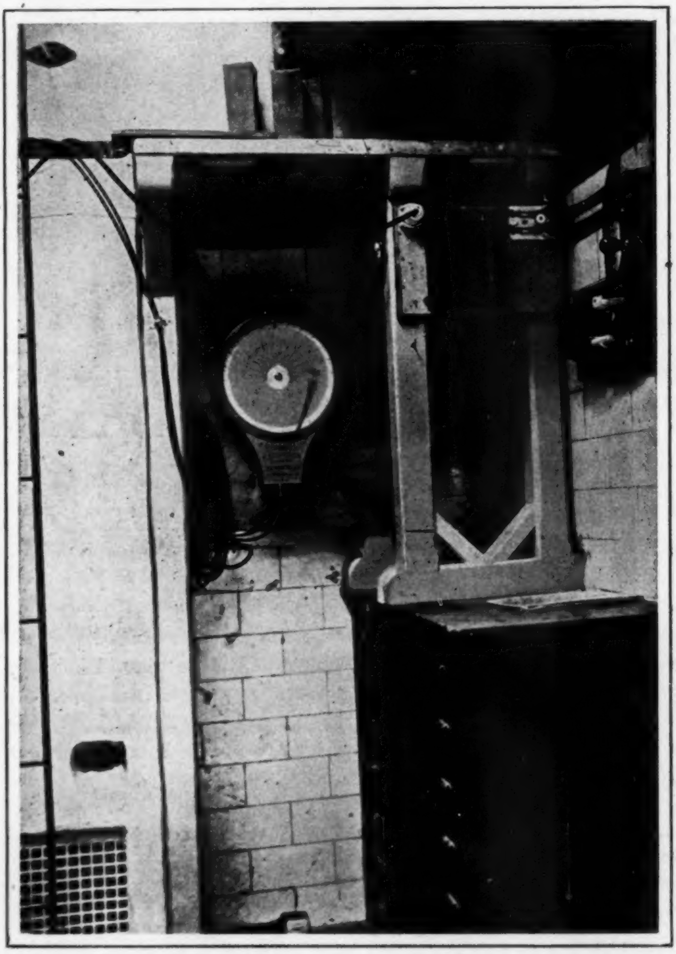


Fig. 8

View in Apparatus Room showing Chart of Recording Psychrometer with hollow gas tubes leading to expansion bulbs in the Observation Room. Just above and to the right of the Recorder is a resistance box controlling the current to the small motor and fan which keeps a current of air moving past the bulbs. On the shelf over the Recorder is located the Exhaust Blower.

thus about 6 feet from the floor. The recorder was placed in the Apparatus Room, where it could be watched by the engineer in charge of the plant. (See Fig. 8.) The manipulation of the plant was simplified enormously by this change. Heretofore, readings were taken at the window each half hour. If the temperature started to change immediately after a reading it was not discovered until the next reading. With the recording chart every little change was apparent at once and proper steps could be taken to counteract the change before it had progressed far. The chart furthermore showed the temperature at the center of the room a much more representative point than at the Observation Room window. From this date on, therefore, a continuous record of wet and dry bulb temperatures was available.

The necessity for air impingement on the wet bulb is indicated by Fig. 9. Numerous check readings with a sling psychrometer

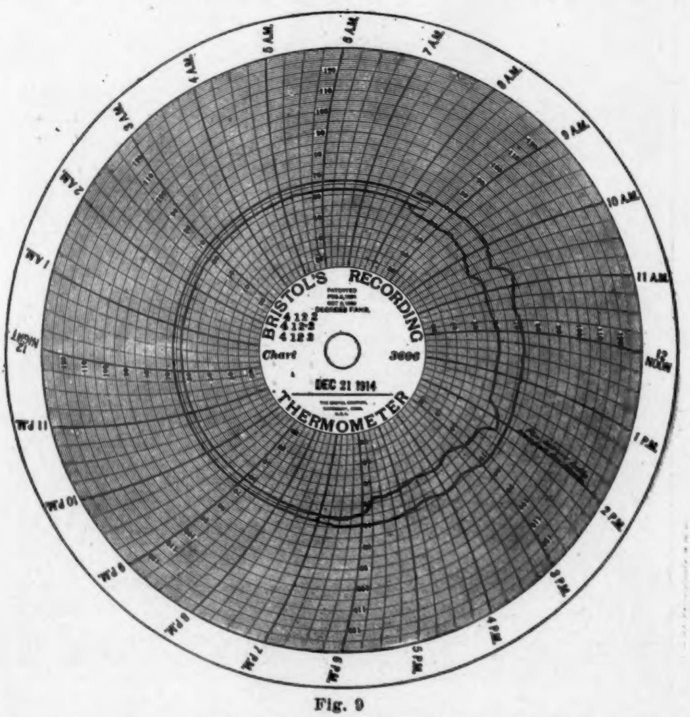


Fig. 9

Air Circulation past Thermometer Bulbs was stopped from 1.45 to 2.15 P. M. The Wet Bulb Temperature rose 8 degrees as a result. The rise would probably have been only half this amount if the wet bulb had been freely exposed to the air of the room.

have shown the recording psychrometer to be very accurate and reliable.

AIR MEASUREMENT

The amount of air forced into the Observation Room was determined by an air meter specially designed for the Commission by Wallace and Tiernan of New York City. (See Fig. 10.) Essen-

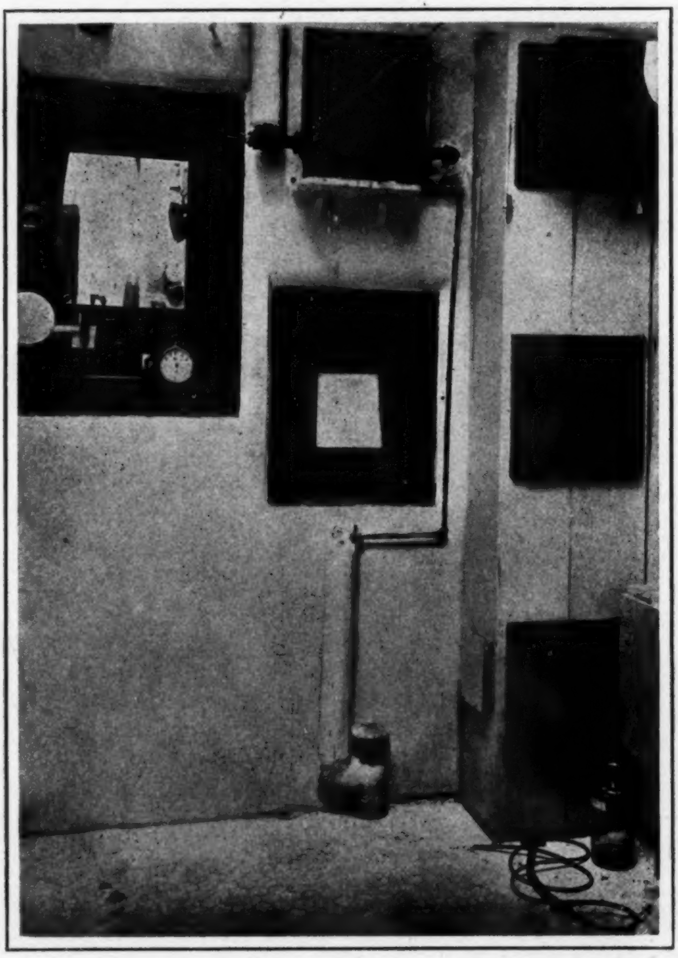


Fig. 10

View in Observation Room, looking toward windows to Apparatus Room. In the upper right corner is the Air Meter attached to supply duct at third opening from floor. The small heater is shown in the upper part of picture.

tially, the meter consists of a tight wooden box one foot square, fitted snugly over the duct opening. The face of the box has a triple wall through which extend from the interior to the exterior 49 brass tubes, 1 inch in diameter and 1⅛ inches in length. Small openings in the inner partition communicate the total pressure of the air flow to the space between the inner and middle partition (See Fig. 11), and this space is connected with one end of the manometer. Openings in the outer partition lead to a space between the outer and middle partitions which is connected to the other end of the manometer registering room pressure. By noting the responses in the liquid of the manometer when passing known quantities of air from a gasometer through the individual brass tubes it has been possible to construct a scale reading in cubic feet per minute. This reading multiplied by the number of holes open represents the total flow of air through the meter in cubic feet per minute.

Several check readings were made against an anemometer, nine positions in the one foot square area being measured. The anemometer values were 17 per cent. higher than the air meter, and in view of the method of calibration, the meter is believed to give the most trustworthy result. There are no ducts in the system of sufficient length to permit a reliable third check reading with a Pitot tube.

The maximum efficiency of the blowers as measured at the air

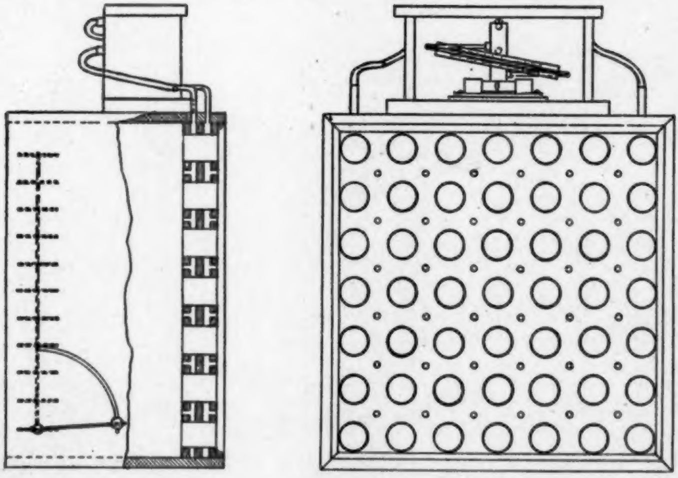


Fig. 11
End Section and Front View of Wallace and Tiernan Air Meter.

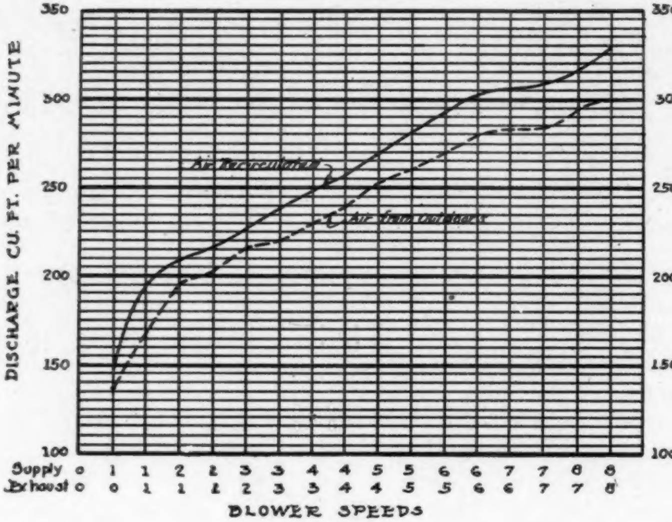


Fig. 12
Output of Blowers at Different Speed Settings.

meter is about 300 cubic feet per minute. This reduction from the rated value of 800 cubic feet per minute is accounted for by the resistance of the conditioning duct.

The air discharge with the various blower settings is illustrated in Table 5 and in Fig. 12.

TABLE 5
Air Discharge through Air Meter at Different Blower Speeds

Speed of Blowers		Recirculated Air		Outdoor Air	
Supply Fan	Exhaust Fan	Manometer Scale	Discharge cu. ft. per minute	Manometer Scale	Discharge cu. ft. per minute
1st	0	About 3.0	147.	About 2.8	137.
1	1	4.0	196.	3.4	167.
2	1	4.3	210.	4.0	196.
2	2	4.4	216.	4.2	203.
3	2	4.65	228.	4.4	216.
3	3	4.85	238.	4.5	220.
4	3	5.1	250.	4.7	230.
4	4	5.25	257.	4.9	240.
5	4	5.5	270.	5.2	254.
5	5	5.75	282.	5.3	260.
6	5	6.0	294.	5.5	270.
6	6	6.2	304.	5.7	280.
7	6	6.25	306.	5.8	284.
7	7	6.3	308.	5.8	284.
8	7	6.5	318.	6.05	296.
8	8	6.7	328.	6.1	299.

AIR CIRCULATION

No serious attempt has been made to study air circulation in the Observation Room, for in an experimental room of this character such data could have but little practical bearing. On one occa-

sion the air stream was traced around the room with an anemometer, its course being as indicated in Fig. 13. The necessity for breaking

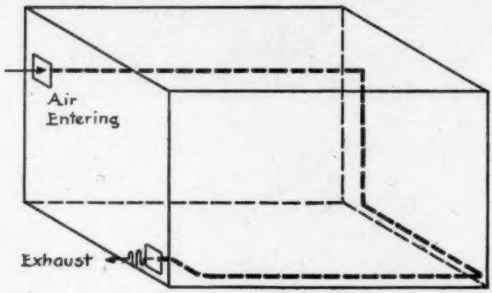


Fig. 13

Course of air current traced with an anemometer from entrance duct in Observation Room to exhaust opening. Air stream cooler than air of room. Discharge 225 cu. ft. per minute, volume of room 1000 cu. ft.

up the entering air column in order to get effective distribution is shown in Table 6 by the high CO_2 readings on days of ample air

TABLE 6

Improvement in Air Distribution Effected by Breaking up Air Stream at Inlet (Room Temp. 68°)

Date	Air Supply cu. ft. per min. per capita.	Theoretical time necessary to completely flush room	CO_2, Parts per 10,000, 2 hours and 45 minutes after 8 subjects entered
		Before Diffusing Entering Air	
12/8	45	3 minutes	20.0
12/10	45	3 minutes	17.0
12/14	45	3 minutes	11.5
		After Diffusing Entering Air	
12/16	45	3 minutes	6.5
12/18	45	3 minutes	6.0

Note: Seven subjects on the 14th, 16th and 18th.

supply. Even with 45 cubic feet per minute per capita it is possible for this column of air to pass on around the room and out the exhaust registers without removing the accumulation of breathed air in the center of the room. Diffusion is favored materially by spreading the air stream as it emerges from the inlet duct, even at the expense of reducing its velocity.

For practical studies in air circulation, this experimental room is further rendered unsuitable by the presence of the cooling coils

at the ceiling in three sides of the room. The dropping of cold air from this location effects the course of air currents or produces counter currents in a way that is quite difficult to measure.

CARBON DIOXIDE DETERMINATIONS

Carbon dioxide analyses are made with the Graham-Rogers modification of the Petterson-Palmquist apparatus, in which a definite air volume is measured before and after absorption in caustic potash solution.

Samples are collected by aspirating air through a pipe extending from the side wall about the center of the Observation Room through the wall into the Apparatus Room. The air is aspirated continuously through a wash bottle of sulphuric acid thus avoiding stagnation in the pipe and insuring a sample representative of the time of collection. The measuring pipette is filled and emptied once with the air to be analyzed before the sample is collected. A double washing in the caustic is made before taking the final reading.

An inspection of the chronological summary on pages 46-49 shows occasional wide variations in the CO_2 results, at times when the room conditions are apparently identical. This is only to be expected, however, on taking all of the facts into consideration. Conditions which influence this variation may be summed up as follows:

1. Air currents past the sampling point induced by blower and fan movement and by temperature.
2. Number of people in room.

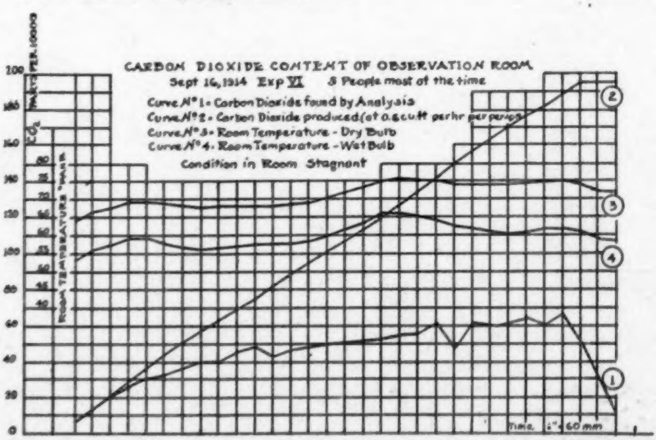


Fig. 14

3. Passing in and out of the room.
4. Leakage depending on differences in temperature and air movement within and without the room.
5. State of activity of subjects.

In Table 7 is given a list of CO_2 results obtained from samples at different levels in the room. Table 8 shows CO_2 readings at various points.

The relation between the production of CO_2 and its retention in the room is graphically illustrated in Fig. 14. A greater portion of that produced escapes. This loss is undoubtedly due to leakage through very small crevices and to diffusion on opening doors and windows. The rapidity of diffusion is best illustrated at the end of the afternoon when the doors were opened and the subjects left the room. The drop from 50 parts at 3.30 p. m. to 10 parts between 4:00 and 4:30 p. m. occurred naturally, no mechanical agitation of the air being applied.

TABLE 7

CO_2 Distribution in Observation Room (Five People in Stagnant Room)

Date	Time	CO_2 Parts per 10,000 — Location of Sampling Point: 1 ft. from floor	6 ft. from floor	9 ft. from floor	Dry Bulb Temp.	Rel. Hum. %
Oct. 9th	11:00 A. M.	37.0	35.0	27.0	68	50
	3:00 P. M.	53.0	52.0	51.0	67.5	52
Oct. 23rd	11:30 A. M.	41.0	23.5	32.5	68.5	53
Oct. 16th	4:00 P. M.	48.0	55.5	40.0	73.	46
Oct. 22nd*	12:00 M.	44.0	44.0	47.0	79.5	64
Nov. 3rd**	3:45 P. M.	87.0	99.0	85.0	74.	47

* No ice in coils nor heat in radiators.
** Heat on coils and skylight.

TABLE 8

CO_2 Distribution in Observation Room on Dec. 17th (Temp. 68°, Rel. Hum. 50%, no air supplied. 7 subjects entered room at 12 noon. Air stirred only by 3 electric fans on shelf 7 ft. above floor.)

Time	Note	CO_2 Parts per 10,000
12:25	Door opened and closed. Sample taken at west wall, center, 5 ft. from floor	53.
12:40	Door opened and closed	
12:42	Sample taken in northeast corner, 5 feet from floor	30.
12:49	Sample taken in northwest corner, 5 feet from floor	37.
12:50	Door opened and closed	
1:00	Sample taken at west wall, center. 9 feet from floor	42.5
1:02	Door opened and closed	
1:19	Sample taken at west wall. center, 1 foot from floor	34.
1:21	Window opened and closed	
1:30	Sample taken at west wall, center, 5 feet from floor	35.

OPERATION DURING AUTUMN MONTHS

A further tax was placed on the flexibility of the plant in the latter part of 1914, the nature of the experiments then being conducted

calling for sudden changes from one air condition to another. In one experiment, 68 degrees and 75 degrees temperature with 50 per cent. relative humidity and no air movement were alternated morning and afternoon, one hour being allowed for the change at noon time. Some difficulty was encountered in raising from 68 degrees to 75 degrees without stopping the cooling plant, the source of heat being a small wall radiator. In view of even more extreme changes anticipated an additional heater, 28 inches by 40 inches, was placed in the room. The effectiveness of this additional heat, is illustrated by Fig. 15, it being desired on this day to run at 40 degrees to 50 degrees from 8 to 11 a. m., at 68 degrees from 12 to 3, and at 40 degrees to 50 degrees from 4 to 6.

As shown in the chronological summary the control of conditions in the Observation Room during this period was much more uniform than in the early part of the work.

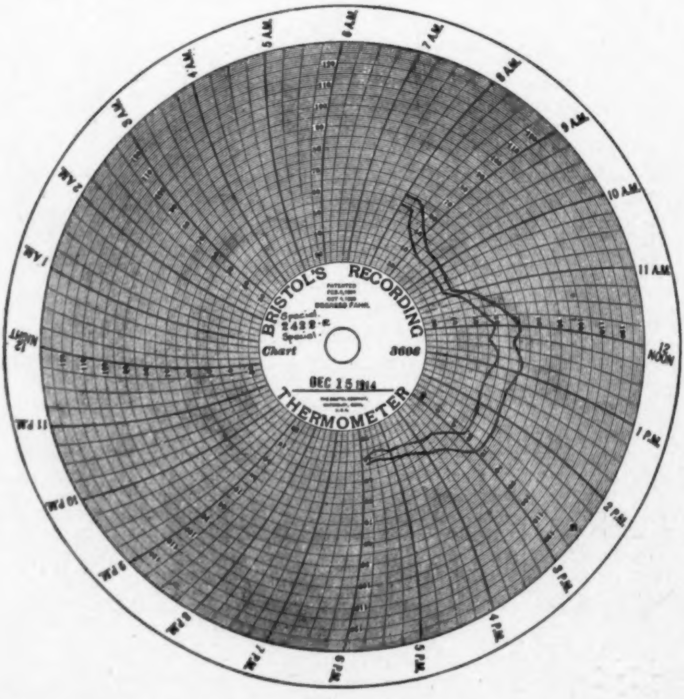


Fig. 15

Psychrometer Chart illustrating Temperature Extremes attainable in Observation Room within a short interval of time.

CONTROL OF CONDITIONS IN APPARATUS ROOM

The original installation in the Apparatus Room provided for separate treatment of air so that both rooms could be used for experiments at the same time. This was found impractical during the early part of the work. Recently, however, both rooms have been put to use by controlling the Apparatus Room with moving air through the washer and the Observation Room by the units within the room itself, no air being supplied.

SPECIAL APPARATUS.

In addition to the equipment already described there have also been used special devices in different phases of the experimental work.

DUST CAGES

Studies on the influence of dust in increasing susceptibility to tuberculosis have required the use of animal cases in which dust is kept constantly in suspension in the air. The illustration, Fig. 16, shows two cages built especially for this purpose by Wallace & Tiernan of New York City. The outer cage or box consists of an iron skeleton framework with tight glass sides which revolves about a wire mesh

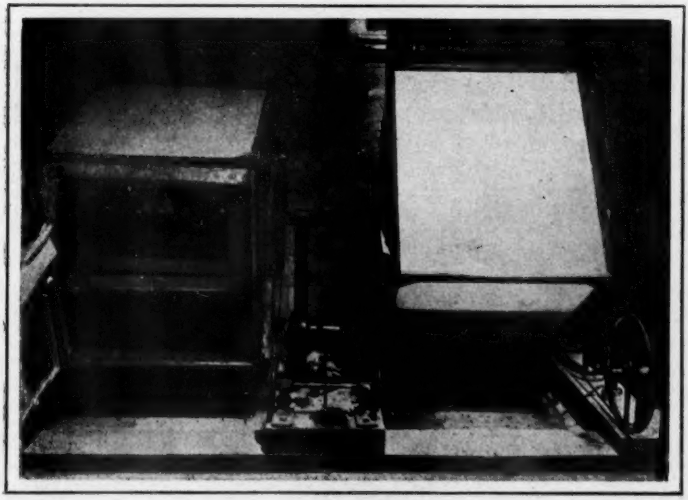


Fig. 16

View of Dust Cages. The wire cage with the animals is located on a stationary platform, about which on a horizontal axis revolves the glass-sided outer cage. Dust is placed in this outer cage and at each revolution is carried up and dropped, thus keeping the atmosphere constantly filled with floating particles. Air is supplied to the animals through the hollow shafts by means of small blower shown in the foreground.

animal cage on a stationary platform. This rotary action keeps dust constantly in the air, carrying it up until the glass side approaches the vertical position and then dropping it to the next glass side. The cages are kept in motion by a small motor with belt and pulley connections. An additional motor and fan supplies air through hollow shafts, the outlet being protected with a cloth filter.

BICYCLE ERGOMETER

The physiological experiments have included studies on work output. For this purpose a bicycle ergometer of the Krogh pattern has been constructed under the direction of Dr. Lee of the Commission (see Fig. 17). This consists of an ordinary bicycle frame with pedals and sprockets. The back sprocket is directly connected to a brass disc about 18 inches in diameter. By means of induction coils and a rheostat variable friction is applied against which the rider must work. On a lever arm extending from the induction coil support may be placed a known weight. The number of revolutions of the disc is recorded on a cyclometer. The weight and the number of the revolutions indicate the exact foot-pounds of work accomplished.

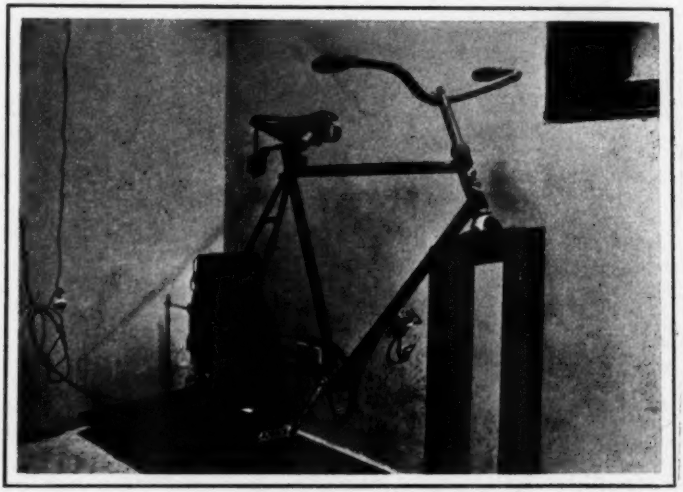


Fig. 17

Corner of Observation Room showing Bicycle Ergometer. This device is fitted with an electrical resistance controlled thorugh a rheostat, and a recorder, so as to give an exact measure of the foot pounds of work accomplished.

BODY TEMPERATURE RECORDER

From the Leeds & Northrup Company of Philadelphia was secured a temperature recorder operated by changes in electrical resistance. This device was placed on the wall of the Observation Room where it has been used to give a continuous record of body temperature over an extended period of time (see Fig. 18). The

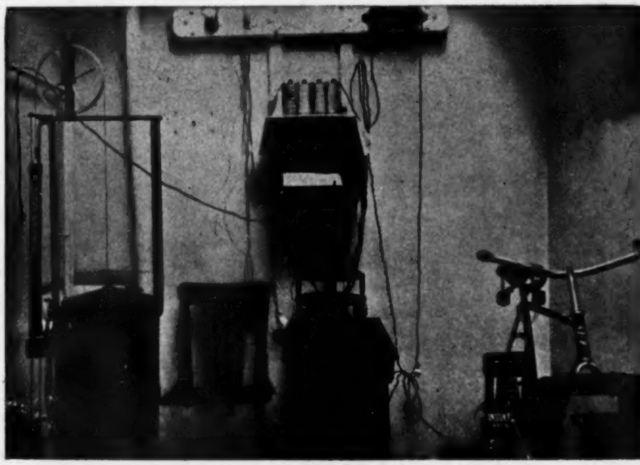


Fig. 18

View in Observation Room looking toward south wall. In the upper center of the picture is the Body Temperature Recorder. In the lower right corner is the Bicycle Ergometer, the controlling rheostat being located on the table. At the left is seen a spirometer used to collect and measure exhaled air.

thermometer bulb is connected to a galvanometer. As the temperature changes the needle is deflected causing a readjustment in the position of a travelling Wheatstone bridge. The recording pen is moved in consequence, travelling on a horizontal track marked out in degrees Centigrade. A small motor operates the mechanism and turns out the co-ordinate paper with the inked record. The use of this device has been somewhat restricted by numerous operating difficulties.

SUMMARY OF AIR CONDITIONS MAINTAINED IN OBSERVATION ROOM, DECEMBER 9, 1913, TO DECEMBER 5, 1914

In Table 9 is presented a chronological summary of the various air conditions maintained in the experiments during the past year.

TABLE 9

DAILY SUMMARY OF PHYSICAL CONDITIONS MAINTAINED IN OBSERVATION ROOM DURING THE VARIOUS EXPERIMENTS FROM DECEMBER 8, 1913, to DECEMBER 5, 1914.

Date	Length of Period	Dry Bulb Temp.			Wet Bulb Temp.			Relative Humidity			Air Supply	CO_2 parts per 10,000		
		Aver.	Max.	Min.	Aver.	Max.	Min.	Aver.	Max.	Min.		Beg.	Mid.	End.
SERIES I. Squads of 4 at rest.														
Condition desired: 86° D.B., 81° W.B., 80% R.H. 45 cu. ft. p.m. p.p. air supply.														
1913														
Dec. 11	2:30-6	86.	87.	81.5	80.5	82.5	77.	80	86	75	50			
17	2:30-6	87.5	89.5	86.5	81.	83.	79.	76	82	62	50			
1914														
Jan. 30	2:30-6	86.5	87.	86.	78.5	80.	75.	78	80	60	55			
Feb. 2	2:30-6	86.5	88.	85.	80.5	81.	79.	76	80	72	45			
10	2:30-6	85.	86.	81.	72.	80.	58.	78	84	53	20			
19	2:30-6	85.5	86.	84.	78.	81.	75.	72	80	60	45		5.5	5.5
25	2:30-6	87.	88.	86.5	76.5	83.	78.	73	81	69	45			6.0
Mar. 6	2:30-6	86.	87.	85.	80.5	82.	76.	78	83	71	45			12.0
9	2:30-6	86.5	87.5	86.	80.5	83.	78.	77	81	70	45	6.0		4.5
16			Omit.											
26	2:30-6	84.5	86.	82.	80.5	82.	79.	84	90	82	45			
Condition desired: 86° D.B., 81° W.B, 80% R.H. No air supply.														
1913														
Dec. 9	2:30-6	83.	87.	80.	74.	80.	70.	65	80	53	0			
15	2:30-6	86.	87.5	82.5	81.	82.5	77.5	80	86	77	50R			
1914														
Jan. 28	2:30-6	88.	88.	87.5	79.5	81.	78.	65	77	57	40R			
Feb. 5	2:30-6	86.5	87.	83.	78.	81.	69.	70	80	49	50R			
13	2:30-6	85.5	86.	84.	80.	81.	78.	80	84	75	50R			
17	2:30-6	86.5	87.	86.	79.	81.	79.	76	79	73	50R			
23	2:30-6	87.	88.	86.5	80.5	83.	78.	75	81	70	45R		5.5	7.5
Mar. 4	2:30-6	88.5	90.	87.5	80.5	82.	78.	70	73	64	40R			7.0
12	2:30-6	87.5	88.5	87.	80.5	82.	80.	74	75	72	40R			
18	2:30-6	85.5	86.5	84.	79.	82.	75.	76	81	66	40R			6.5
27	2:30-6	85.5	86.	83.5	80.5	82.	77.	81	88	81	40R	6.0		
Condition desired 86° D.B., 81° W.B., 80% R.H. No air supply. Air movement within room created by electric fans														
1913														
Dec. 10	2:30-6	86.	89.	81.	77.	84.	67.	70	83	48	50R			
16	2:30-6	87.5	88.5	85.	81.	82.5	79.	77	82	68	50R			
1914														
Jan. 29	2:30-6	88.	88.	87.5	80.5	81.5	80.	74	76	70	55R			
Feb. 6	2:30-6	86.5	88.	86.	80.	82.	77.	72	77	66	50R			
9	2:30-6	88.5	91.	87.5	78.5	84.	77.	68	74	56	45R			
18	2:30-6	87.	87.5	85.5	80.	81.	73.	74	77	70	40R	6.0	8.5	
24	2:30-6	86.5	87.5	83.5	79.5	81.	64.	74	79	58	43R			8.0
Mar. 5	2:30-6	88.5	89.5	85.5	73.5	78.	71.	47	68	46	0		16.0	26.0
13	2:30-6	88.	89.	86.	79.5	82.	78.	68	79	64	30R	4.5		9.0
17	2:30-6	86.	87.	85.	72.	78.	68.	50	63	40	40R			
25	2:30-6	84.5	85.5	82.	78.	81.	71.	75	80	58	43R	4.5		8.5
Condition desired: 68° D.B., 57° W.B., 50% R.H., 45 cu. ft. p.m. p.p. air supply.														
1913														
Dec. 8	2:30-6	69.	71.5	68.	58.5	60.	57.	53	57	50	45			
19	2:30-6	68.	70.	67.	57.	60.	53.	48	62	38	50			
1914														
Jan. 27	2:30-6	66.5	67.5	64.5	58.	58.5	56.5	58	61	54	55			
Feb. 4	2:30-6	67.5	68.	66.	54.	55.	53.	42	49	38	45			
12	2:30-6	67.	71.	64.	55.	59.	52.	44	51	38	45			
16	2:30-6	68.	70.	66.5	57.5	62.	54.	52	63	41	45			
27	2:30-6	68.5	69.5	66.5	57.	62.	54.	47	51	37	45			8.0
Mar. 3	2:30-6	68.	69.	68.	56.5	57.	57.	48	50	46	45			5.0
11	2:30-6	68.5	70.	66.5	57.	59.	56.	51	55	48	45	6.5		6.0
20	2:30-6	67.5	68.	65.5	57.5	57.	57.	54	59	48	45	6.0		6.5
23	2:30-6	67.5	68.	66.5	57.5	59.	54.	54	66	50	45			

TABLE 9—Continued

Date	Length of Period	Dry Bulb Temp.			Wet Bulb Temp.			Relative Humidity			Air Supply	CO_2 parts per 10,000		
		Aver.	Max.	Min.	Aver.	Max.	Min.	Aver.	Max.	Min.		Beg.	Mid.	End.
Condition desired: 68° D.B., 57° W.B., 50% R. H. No air supply.														
1913														
Dec. 12	2:30-6	71.5	72.5	71.	56.	56.	56.	36	38	34	0			
18	2:30-6	74.	75.5	71.	58.	61.	54.	36	40	31	0			
1914														
Jan. 26		Experiment Room not used.												
Feb. 3	2:30-6	72.5	75.	70.	60.5	62.	59.	52	52	45	45R			
11	2:30-6	69.	71.5	65.	59.	61.	58.	48	53	28	50R			
20	2:30-6	74.	74.	72.5	57.5	59.	56.	35	40	30	20R	6.5		
26	2:30-6	67.5	68.5	65.	57.	58.	54.	52	57	48	40R			9.0
Mar. 2	2:30-6	67.	68.5	66.	57.	58.	55.	54	57	45	40R		12.0	7.0
10	2:30-6	68.	70.	64.	57.5	60.	54.	53	62	47	40R			7.5
19	2:30-6	70.	72.	68.5	56.5	60.	55.	43	52	38	40R			
24	2:30-6	69.	71.	65.5	56.5	58.	54.	45	53	40	35R			
SERIES II. Squads of 4 at rest.														
Condition desired: 68° D.B., 57° W.B., 50% R.H. 45 c.f. p.m. p.p. Air supply.														
1914														
Mar. 30	2:30-6	68.	68.5	66.5	57.	59.	56.	51	54	50	45			
31	2:30-6	68.5	69.	66.	57.	57.	56.	48	52	43	45			
Apr. 1	2:30-6	67.5	68.5	67.	57.	58.	57.	52	53	50	45	4.0		5.0
2	2:30-6	67.5	69.5	64.	55.5	64.	53.	46	52	41	45			
3	2.30-6	68.0	68.5	67.5	57.	58.	56.	50	54	45	45	4.0		5.0
Condition desired: 86° D.B., 81° W.B., 80% R.H. No air supply.														
6	2:30-6	86.5	87.	85.	80.	82.	73.	76	80	68	45			
7	2:30-6	87.	87.5	86.5	79.5	81.	78.	72	75	67	45	5.0		7.0
8	2:30-6	86.	86.	85.	81.	81.	80.	79	80	79	45	5.0		7.5
9	2:30-6	86.	86.5	85.5	80.5	82.	79.	78	80	73	45			6.0
10	2:30-6	85.5	86.	85.	80.	82.	79.	79	83	77	45			7.0
Condition desired: 86° D.B., 81° W.B., 80% R.H. No air supply.														
Apr. 13	2:30-6	86.5	87.	86.	80.	80.	79.	71	75	68	0	4.5		10.5
14	2:30-6	86.	86.5	85.	81.	81.	79.	78	80	75	0			8.0
15	2:30-6	86.	86.5	84.5	81.	82.	77.	78	83	75	0	4.5		5.5
16	2:30-6	85.5	86.	84.	80.5	82.	78.	81	82	75	0	3.0		7.0
17	2:30-6	86.	87.	85.5	80.5	83.	80.	79	84	77	0	2.5		8.[illegible]
Condition desired: 68° D.B., 57° W.B., 50% R.H. 45 cu. ft. p.m. p.p. air supply.														
Apr. 20	2:30-6	69.5	72.5	67.	60.	63.	55.	47	49	44	45	2.		5.
21	2:30-6	68.	69.	65.	56.	58.	57.	58	62	53	45	3.		5.
22	2:30-6	69.	70.5	67.	58.5	60.	58.	53	59	47	45	4.		
23	2:30-6	70.	71.	69.	58.	59.	57.	48	51	45	45	4.		5.5
24	2:30-6	69.	70.5	67.5	57.5	59.	56.	48	60	41	45	4.5		
May 4	2:30-6	69.	72.5	66.5	57.5	60.	56.	48	50	47	45			
5	2:30-6	67.5	70.	65.5	59.	61.	58.	61	67	64	45	5.		6.5
6	2:30-6	68.	69.5	65.5	57.	60.	55.	50	57	48	45	4.		7.
7	2:30-6	67.	71.	64.5	56.5	61.	53.	52	59	42	45	3.5		5.
8	2:30-6	67.5	71.	64.	58.	62.	54.	56	65	52	45			13.5
Condition desired: 86° D.B., 81° W.B., 80% R.H. No air supply.														
May 11	2:30-6	88.5	92.	86.5	82.	92.	87.	76	96	65	0	4.		6.
12	2:30-6	84.	85.	81.	79.	82.	75.	77	83	69	0	7.		8.
13	2:30-6	86.5	90.	82.	81.5	87.	68.	81	87	75	0	7.5		24.
14	2:30-6	87.	94.	84.	80.	82.	69.	74	84	36	0	8.		26.
15	2:30-6	86.5	88.	85.5	81.5	83.	64.	81	86	77	0	3.		77.
18	2:30-6	86.	86.5	84.5	81.	83.	77.	81	84	77	0	5.		42.
19	2:30-6	85.5	86.5	82.5	80.	82.	69.	81	82	78	0	8.		45.
20	2:30-6	86.	86.	84.5	81.	82.	75.	81	83	80	0	6.		44.
21	2:30-6	85.5	86.	84.5	80.5	82.	79.	81	83	72	0	6.		29.
22	2:30-6	86.5	88.	84.	81.	84.	79.	81	88	78	0	5.5		29.5

TABLE 9—Continued

Date	Length of Period	Dry Bulb Temp. Aver.	Dry Bulb Temp. Max.	Dry Bulb Temp. Min.	Wet Bulb Temp. Aver.	Wet Bulb Temp. Max.	Wet Bulb Temp. Min.	Relative Humidity Aver.	Relative Humidity Max.	Relative Humidity Min.	Air Supply	CO_2 parts per 10,000	CO_2 parts per 10,000 Beg.	CO_2 parts per 10,000 Mid.	CO_2 parts per 10,000 End.
Condition desired: 68° D.B., 57° W.B., 50% R.H. 45 cu. ft. p.m. p.p. air supply.															
May 25	2:30-6	68.5	69.5	67.5	58.	59.	57.	52	54	49	45		6.		5.
26	2:30-6	68..	69.	66.	56.5	58.	56.	52	54	45	45		3.5		5.
27	2:30-6	67.5	70.	65.	58.	60.	55.	54	57	50	45		4.5		5.5
28	2:30-6	68.	69.	67.	57..	58.	56.	49	50	47	45		4.		5.
29	2:30-6	68.	69.	67.5	57..	58.	56.	50	52	46	45		5.		5.5
SERIES III. Squads of 4 at rest and at work (last 2 days of week).															
Condition desired: 68° D.B., 57° W.B., 50% R.H. 45 cu. ft. p.m. p.p. air supply.															
June 8	1:30-4	68..	70.	66.	60..	62.	59.	63	66	62	45				6.5
9	8:30-4	68.	70.	67.	57.	68.	56.	51	54	48	45		5.5	6.	
10	8:30-4	70.	73.	65.	61.	65.	57.	57	58	48	45		6.	4.5	5.
11	8:30-3:30	67..	72.5	66.	58.	60.	57.	57	58	45	45		6.	4.5	5.
12	8:30-3:30	68.	73.	64.	60.	65.	60.	62	70	53	45		6.	6.	
Condition desired: 86° D.B.. 81° W.B., 80% R.H. No air supply.												9:00	11:30	2:00	3:30
June 15	9-4	86.	87.5	83.5	79.5	83.	72.	76	86	55	0	19.		31.	
16	8:30-3:30	86.5	89.5	84.	81.	82.	78.	79	82	76	0	4.5	30.5		29.
17	8:30-4:00	87..	93.	83.	79.	83.	70.	71	83	54	0	10.	8.5		29.5
18	8:30-4:00	85..	87.5	82.5	80.	83.	77.	81	85	78	0	9.5	21.	29.	17.5
19	8:30-3:30	86.	87.5	82.5	81.	83.	79.	82	86	78	0	11.5	31.	36.5	29.
Condition desired: 68° D.B., 57° W.B., 50% R.H. No air supply.															
June 22	8:30-3:30	69.5	74.	66.5	59.	63.	57.	54	68	47	0	8.5	31.	40.5	44.
23	9-3:30	67.5	70.	65.	57.	58.	54.	50	56	46	0		25.	39.5	32.5
24	8:30-3:30	67.5	70.5	64.5	57.	59.	54.	51	58	48	0	15.5	23.5	31.	32.
25	9:30-4	66.5	69.	64.	57.	59.	54.	54	63	49	0	11.	57.	54.	44.5
26	9:30-4	67.5	69.5	66.	57.	59.	54.	52	55	43	0	15.5	42.	51.	47.
Condition desired: 86° D.B.. 81° W.B., 80% R.H. 45 cu. ft. p.m. p.p. air supply.															
June 29	10-4	84.	87.5	78.5	76.	80.	69.	70	78	60	45		12.	6.5	6.
30	9-4	86.	89.	85.	81.	82.	79.	79	84	74	45	7.5	7.	8.5	5.
July 1	9-4	86.5	87.5	85.	81.	82.	80.	80	84	74	45	5.5	9.	5.	4.
2	9:30-3:30	86.5	90.	84.	81.	82.	80.	80	85	75	45	5.5	10.5	5.	
3	Experiment omitted.														
Condition desired: 75° D.B., 62.5° W.B., 50% R.H. 45 c.f. p.m. p.p. air supply.															
July 6	9:30-4	74.	79.5	72.	63.	69.	60.	54	65	47	45	6.	5.5	9.	7.
7	9:30-3:30	75.	82.	70.	64.	69.	61.	56	59	50	45	4.5	8.		9.
8	9-4	75..	84..	67.5	66.	72.	66.	63	73	55	45	7.	18.	6.5	6.
9	9-4	81.	86.5	74.	73.	79.	66.	68	77	60	45	8.	17.5	8.5	7.
10	9-4	81.	81.	74.	72.	77.	66.	66	71	61	45	5.	21.	7.5	7.
Condition desired: 86° D.B.. 81° W.B.. 80% R.H. No air supply, 5 desk fans.															
July 13	9-4	91.5	96	86	85	91.	77	76	87	55	0				19.
14	9-4	90.	93	84	84	88.	79	79	87	68	0	4.	11.	19.	19.
15	9-4	87.5	92	85	82.5	86.	80	80	84	71	0	5.	18.	15.	20.
16	9-4	86.	91	84	79	82.	72	74	84	48	0	7.	35.	29.5	33.
17	9-4	85.5	87	84	81	83.	79	81	85	70	0	3.5	32.	34.5	30.
SERIES IV. Squad of 2 at Rest and at Work.															
Condition desired; 86° D.B., 81° W.B., 80% R.H. 45 c.f. p.m. p.p. air supply.															
July 27	9-4	86.5	91	85	80.5	81.	79	79	83	61	115			8..	5.
28	9-4	85.5	88	84	79.5	82.	77	79	86	66	75	5.5	5.	5.5	5.
29	9-4	86.	89	85	81	81.	79	81	92	73	75	7.5	7.	6.5	7.
30	9-4	85.5	87	84	81	82.	79	81	84	74	75	6.	5.5	6..	6.5
31	9-4	86.5	89	86	81	82.	80	80	86	75	45	6.	5.5	7.5	6.
Condition desired: 68° D.B., 57° W.B., 50% R.H. No air supply.															
Aug. 3	9-4	66	75	66	59	61.	56	54	63	42	0	17.	26.		37.5
4	9-4	68	70	66	58	60.	57	56	61	50	0	15.5	18.	16.5	17.
5	9-4	68	70	67	57.5	60.	57	53	58	50	0	15.	33.	32..	35.
6	9-4	68	71	66	57	60.	55	50	54	48	0	15.5	34.5	35..	33.5
7	9-4	67.5	70	66.	56.5	60.	55	52	57	48	0	12.5	23.	24.5	24.

TABLE 9—Continued

Date	Length of Period	Dry Bulb Temp. Aver.	Dry Bulb Temp. Max.	Dry Bulb Temp. Min.	Wet Bulb Temp. Aver.	Wet Bulb Temp. Max.	Wet Bulb Temp. Min.	Relative Humidity Aver.	Relative Humidity Max.	Relative Humidity Min.	Air Supply	CO_2 parts per 10,000 9.00	11.30	2.00	3.30
Condition desired: 75° D.B., 62.5 W.B., 50% R.H. 45 c.f. p.m. p.p. air supply.															
Aug. 10	9-4	78.5	82	75	69.5	73.	66	66	70	45	45	5.5	4.5	8.	5.
11	9-4	75	80	73	66.5	72.	65	64	68	45	45	6.5	5.	5.5	
12	9-4	75.5	79	74	67.5	70.	66	65	69	50	45	5.	5.	5.	5.
13	9-4	75	76	73	61	64.	59	46	50	45	45	5.5	4.	4.5	5.
14	9-4	Experiment omitted.													
Condition desired: 75° D.B., 62.5° W.B., 50% R.H. No air supply.															
Aug. 24	9-4	75	83	74	62.	63.	62	50	53	47	0	27.	31.	38.	33.
25	9-4	75.5	77	75	62.5	64.	61	49	53	41	0	16.	17.	19.	10.
26	9-4	75.5	78	74	63	66.	61	50	56	45	0	13.5	9.	17.5	
27	9-4	75	77	75	63	65.	61	50	56	46	0	7.	19.5	20.	19.
28	9-4	74.5	77	72	63	65.	61	54	65	45	0	11.5	21.	25.	29.
Condition desired: 86° D.B., 81° W.B., 80% R.H. No air supply.															
Aug. 31	9-4	86	87	84	80.5	82.	77	80	84	73	0	12.	24.	18.5	23.
Sept. 1	9-4	86	89	84	81	82.	79	80	84	77	0	11.	10.	17.5	
2	9-4	85	87	83	82	84.	77	84	94	77	0	21.	22.	17.	18.5
3	9-4	86	87	85	80	83.	78	80	86	72	0	9.5	24.	36.	36.
4	9-4	86	88	86	81	84.	77	80	86	74	0	13.	23.	25.5	28.
SERIES V. Squad of 4 at Rest. 6 Air Conditions.															
Aug. 17	9-4	86	87	84	80	82.	73	79	88	51	45	11.5	4.5	6.5	6.5
18	9-4	75	77	71	63	65.	59	49	52	48	0	12.	26.5	25.5	23.
19	9-4	69	74	67	58	62.	55	51	53	46	0	24.	52.	48.	52.
20	9-4	75	80	73	65	67.	64	59	64	52	45	6.	6.	4.5	5.5
21	9-4	86	87	85	81	80.	83	81	84	79	0	23.	27.	31.	34.5
22	9-4	70	71	68	61	61.	59	57	60	52	45	11.5	10.5	11.	11.
SERIES VI. Squads of 4 at Rest.															
Condition desired: 68° D.B., 57° W.B., 50% R.H. No air supply.															
Sept. 8	9-4	68	71	66	57	61.	55	50	53	45	0	10.5	35.	32.5	36.5
9	9-4	68	69	67	58	59.	56	51	58	45	0	14.5	29.	27.5	27.
10	9-4	68	69	67	57	58.	55	50	54	45	0	13.5	25.5	34.5	44.
11	9-4	68	69	67	57	58.	56	50	54	47	0	12.	22.5	27.5	38.
12	9-4	68	69	67	57	58.	56	50	54	46	0	11.	20.	24.5	32.5
Condition desired: A. M., 68° D.B., 57° W.B., 50% R.H. No air supply. P. M., 75° D.B., 62.5° W.B., 50% R.H. No air supply.															
Sept. 14a	9-12	67.5	69.	67.	57.	58.	56.	50	53	47	0	14.5	21.5		
p	1-4	74.	75.	72.	63.	64.	61.	53	63	48	0	.		22.	32.5
15a	9-12	68.5	70.	68.	58.	60.	57.	51	59	49	0	11.	26.5		
p	1-4	74.5	75.	74.	63.	65.	61.	52	58	48	0	.		48.	37.
16a	9-12	67.5	69.	64.	57.	58.	53.	51	52	47	0	6.5	48.5		
p	1-4	74.5	75.	73	63.	66.	61.	54	63	47	0	.		61.5	51.5
17a	9-12	68.5	70.	67.	57.	58.	55.	50	51	45	0	6.	40.5		
p	1-4	74.5	76.	74.	62.	64.	62.	50	54	47	0	.		76.	68.5
18a	9-12	76.5	77.	76.	63.	63.	63.	50	50	48	0	6.	6.		
p	1-4	74.5	75.	72.	62.	62.	59.	49	50	43	0	.		38.	36.5
Condition desired: A.M., 75° D.B., 62.5° W.B., 50% R.H. No air supply. P. M., 68° D.B., 57° W.B., 50% R.H. No air supply.															
Sept. 21a	9-12	75.	75.	74.	62.	63.	62.	50	51	48	0	6.5	52.		
p	1-4	68.	69.	68.	57.	58.	56.	50	51	46	0	.		66.5	55.5
22a	9-12	74.5	75.	74.	63.	63.	62.	51	53	49	0	10.	51.		
p	1-4	68.	69.	68.	57.	57.	57.	50	50	49	0	.		60.5	55.5
23a	9-12	75.	76.	75.	62.	63.	61.	50	53	47	0	9.5	56.		
p	1-4	68.	69.	68.	57.	58.	56.	49	53	46	0	.		61.	48.
24a	9-12	74.5	75.	73.	62.	64.	61.	50	54	48	0	9.5	43.		
p	1-4	69.5	71.	69.	57.	59.	56.	47	49	41	0			61.	62.
25a	9-12	75.	78.	74.	63.	66.	61.	50	53	44	0	10.5	42.		
p	1-4	68.	69.	67.	57.	58.	55.	50	51	43	0	.		52.	59.5

TABLE 9—Continued

Date	Length of Period	Dry Bulb Temp.			Wet Bulb Temp.			Relative Humidity			Air Supply	CO_2 parts per 10,000			
		Aver.	Max.	Min.	Aver.	Max.	Min.	Aver.	Max.	Max.		9.00	11.30	2.00	3.30
Condition desired Sept. 28, 29, 30, Oct. 8, 9, 10: 68°D.B., 57° W.B., 50% R.H. No air supply. Oct. 1, 2, 3, 5, 6, 7: 75° D.B., 62.5° W.B., 50% R.H. No air supply.															
Sept. 28	9-12	68.	69.	67.	57.	58.	56.	50	52	47	0	7.	48.	53.	66.5
29	1-4	68.	69.	68.	57.	58.	57.	50	51	46	0	10.5	32.	38.	46.
30	9-12	68.	69.	67.	67.	59.	56.	50	55	49	0	17.	49.	63.5	62.5
Oct. 1	1-4	74.	75.	71.	62.	66.	60.	51	65	46	0	19.	45.5	51.	
2	9-12	75.	79.	74.	62.	63.	61.	49	51	41	0	9.	41.	71 5	84.
3	1-4	75.	76.	73.	63.	66.	62.	50	57	47	0		43.	48.	54.
5	9-12	75	76	72.	62.	64.	60.	49	54	47	0	5.	55.	63.	53.
6	1-4	75.	77.	75.	64.	70.	62.	53	53	48	0	8.	32.	34.	39.5
7	9-12	75.	77.	75.	63.	64.	62.	49	53	47	0	8.	39.	51.	53.5
8	1-4	68.	69.	67.	57.	59.	57.	50	55	47	0	10.	38.5	44.	48.
9	9-12	68.	69.	67.	57.	59.	57.	51	58	49	0	6.	35.	53.	52.
10	1-4	68.	69.	67.	57.	61.	56.	51	65	48	0	11.	44.	65.	72.
SERIES VII. Squad of 4 at Work and at Rest.															
Condition desired: 75° D.B., 62.5° W.B., 50% R.H. No air supply.															
Oct. 12	9-4	75.	76.	73.	63.	65.	62	54.	60	48	0	5.5	60.5	85.	90.
16	9-4	75.	76.	73.	63.	68.	66	51.	70	46	0	6.5	42.5	68.5	62.5
22	9-4	76.	80.	74.	65.	71.	61	56.	67	45	0			32.5	43.
28	9-4	75.	77.	74.	64.	68.	61	54.	68	45	0	4.	43.5	38.5	29.5
Nov. 3	9-4	75.	76.	74.0	63.	66.	61	54.	60	47	0	5.	135.	125.	99.
Condition desired: 68° D.B., 57° W.B., 50% R.H. No air supply.															
Oct. 13	9-4	68.	69.	67.	57.5	59.	56	51.	54	50	0	5.	38.	85.	51.
19	9-4	68.	70.	67.	57.	60.	55	50.	57	43	0	10.5	50.	72.5	72.5
23	9-4	68.	70.	66.	57.	59.	55	50.5	58	41	0	4.5	23.5	42.5	52.5
29	9-4	68.	69.	67.	58.	60.	55	52.	63	44	0	10.5	67.5	63.5	51.
Nov. 4	9-4	68.	72.	66.	58.5	59.	56	54.	60	46	0	5.	52.	48.	51.
Condition desired: 75° D.B., 62.5° W.B., 50% R.H. 45 c.f. p.m. p.p. air supply.															
Oct. 14	9-4	75.	78.	72.	62.5	64.	60	49.5	53	43	45	5.5	14.	13.	12.
20	9-4	75.	78.	74.	62.5	65.	61	49.5	62	43	45	6.	8.5	12.5	11.
26	9-4	75.5	77.	74.	63.	64.	61	50.	54	43	45	4.5	8.5	9.	7.5
30	9-4	75.5	81.	75.	63.	65.	61	48.	53	42	45	4.	14.5	16.	10.5
Nov. 5	9-4	75.	77.	74.	62.	64.	60	47.	51	42	45	4.5	8.5	6.	6.5
Condition desired: 68° D.B., 57° W.B., 50% R.H. 45 c.f. p.m. p.p. air supply.															
Oct. 15	9-4	68.5	72.	68.	57.5	61.	57	51.5	54	50	45	10.	16.	12.	10.
21	9-4	68.5	70.	68.	57.5	59.	56	49.5	55	45	45	6.	11.	12.	11.
27	9-4	68.5	72.	67.	57.	58.	55	49.	53	42	45	3.	8.5	10.	9.5
Nov. 2	9-4	68.	69.	67.	57	60.	56	50.5	55	48	45	5.	10.5	15.	6.5
6	9-4	67.5	70.	66.	56.5	60.	56	48.5	56	40	45	5.5	8.3	6.	6.
Conditions desired: Nov. 9, 10, 11, 19, 20, 21—68° D.B., 57° W.B., 50% R.H. No air supply. Nov. 12, 13, 14, 16, 17, 18—75° D.B., 62.5° W.B., 50% R.H. No air supply.															
Nov. 9	9-4	68.5	72.	68.	57.5	70.	57	50.5	56	44	0	11.5	21.5	31.5	29.5
10	9-4	68.	70.	67.	56.5	59.	56	50.	54	47	0	9.	36.5	43.	52.
11	9-4	68.	69.	68.	57.	58.	55	50.	52	45	0	4.5	42.	51.5	55.
12	9-4	75.	76.	72.	62.5	64.	62	49.5	51	47	0	17.5	25.	48.	17.5
13	9-4	75.	76.	74.	62.5	65.	61	49.5	53	45	0	5.5	26.5	26.	11.5
14	9-4	75.	76.	73.	63.	65.	60	51.	56	47	0	17.	35.	53.	47.5
16	9-4	75.	77.	73.	64.5	69.	60	54.	67	46	0	5.5	36.	38.5	48.5
17	9-4	75.	76.	73.	62.5	66.	60	50.5	61	46	0	5.	32.	20.	34.
18	9-4	74.5	76.	71	62.5	66.	58	52.	50	45	0	7.	21.	27.5	31.
19	9-4	68.	70.	68.	57.	59.	57	49.5	55	47	0	13.5	30.	31.5	37.
20	9-4	68.	70.	67.	57.5	58.	57	50.5	54	48	0	14.5	32.	32.	22.
21	9-4	68.5	69.	67.	57.	58.	57	50.	54	48	0	15.	43.	54.	58.

TABLE 9—Continued

Date	Length of Period	Dry Bulb Temp.			Wet Bulb Temp.			Relative Humidity			Air Supply	CO_2 parts per 10,000			
		Aver.	Max.	Min.	Aver.	Max.	Min.	Aver.	Max.	Min.		9.00	11.30	2.00	3.30
Condition desired: A. M.—68° D.B., 57° W.B., 50% R.H. No air supply. P. M.—75° D.B., 62.5° W.B., 50% R.H. No air supply.															
Nov. 23a	9-12	68.5	72.	67.	56.5	60.	56	50.	53	48	0	10.	28.5		
p	1-4	75.	76.	73.	63.	66.	60	50.	51	48	0			38.	46.
24a	9-12	68.	69.	66.	57.	59.	56	49.	51	47	0	10.5	26.		
p	1-4	75.5	78.	74.	63.	64.	60	49.5	55	46	0			38.5	47.
25a	9-12	68.	69.	66.	56.5	57.	55	48.5	49	44	0	8.5	26.		
p	1-4	74.5	76.	74.	62.5	63.	60	51.	61	47	0			34.	39.
27a	9-12	68.	69.	67.	57.	59.	56	49.5	53	44	0	6.	24.		
p	1-4	74.	75.	72.	62.	63.	60	49.	53	44	0			34.	46.
28a	9-12	68.5	70.	68.	57.	59.	54	48.5	51	44	0	10.	41.		
p	1-4	74.5	76.	73.	62.	65.	60	48.5	52	44	0			43.5	48.
Condition desired: A. M.—75° D.B., 62.5° W.B., 50% R.H. No air supply. P. M.—68° D.B., 57° W.B., 50% R.H. No air supply.															
Nov. 30a	9-12	75.	80.	73.	62.	65.	59	48.	53	44	0	7.5	22.5		
p	1-4	68.	69.	67.	57.	59.	56	49.	55	46	0			23.5	25.
Dec. 1a	9-12	75.	77.	73.	63.	66.	61	51.	56	48	0	9.	35.5		
p	1-4	68.	70.	66.	57.	59.	53	48.	51	42	0			34.5	37.
2a	9-12	74.	75.	73.	62.	65.	59	50.	56	44	0	13.	34.		
p	1-4	68.	69.	67.	57.	58.	56	49.5	53	46	0			42.5	41.
4a	9-12	75.	77.	74.	62.	64.	51	47.5	51	45	0	9.	21.		
p	1-4	67.5	70.	66.	55.5	58.	54	46.	50	38	0			23.5	26.
5a	9-12	74.5	76.	72.	62.	65.	59	49.	53	42	0	8.	20.		
p	1-4	68.	70.	48.	57.	58.	55	48.5	50	40	0			23.	

B—REACTIONS OF SUBJECTS IN DIFFERENT AIR ENVIRONMENTS

The experiments of the past year may be divided for convenience into eight distinct periods or series and a brief digest of the results of these studies will be taken up in this order. (Table 10.)

TABLE 10

Series No.	Time	Air Condition Studied
I.	Dec. 8, 1913-Mar. 27, 1914	86 degrees—High humidity—Air supply 86 degrees—High humidity—Air supply—desk fans 68 degrees—Medium humidity—Air supply
II.	Mar. 30-May 29, 1914	86 degrees—High humidity—Air supply 86 degrees—High humidity—Stagnant 68 degrees—Medium humidity—Air supply
III.	June 8-July 17, 1914	86 degrees—High humidity—Air supply 86 degrees—High humidity—Stagnant 86 degrees—High humidity—Stagnant—desk fans 75 degrees—Medium humidity—Air supply 68 degrees—Medium humidity—Air supply 68 degrees—Medium humidity—Stagnant
IV.	July 27-Sept. 4, 1914	86 degrees—High humidity—Stagnant 86 degrees—High humidity—Air supply 75 degrees—Medium humidity—Air supply 75 degrees—Medium humidity—Stagnant 68 degrees—Medium humidity—Stagnant
V.	Aug. 17-22, 1914	86 degrees—High humidity—Stagnant 86 degrees—High humidity—Air supply 75 degrees—Medium humidity—Air supply 75 degrees—Medium humidity—Stagnant 68 degrees—Medium humidity—Stagnant 68 degrees—Medium humidity—Air supply
VI.	Sept. 8-Oct. 10, 1914	75 degrees—Medium humidity—Stagnant 68 degrees—Medium humidity—Stagnant
VII.	Oct. 12-Nov. 6, 1914	75 degrees—Medium humidity—Stagnant 75 degrees—Medium humidity—Air supply 68 degrees—Medium humidity—Air supply 68 degrees—Medium humidity—Stagnant
VIII.	Nov. 9-Dec. 5, 1914	75 degrees—Medium humidity—Stagnant 68 degrees—Medium humidity—Stagnant

NATURE OF TESTS

In determining the relative effects on the human body of different factors in the air environment, three avenues of approach suggest themselves:

1. Measurement of mental accomplishments.
2. Measurement of physiological responses.
3. Recording the opinion of the subject as to state of comfort.

With this object in view the Commission prepared certain definite programs of mental and physical tasks and tried them out under various air conditions.

As a rule a squad of four men has been used as a working unit. These men, for the most part students of the City College and neighboring institutions ranging from 16 to 22 years of age, report for instructions a day or so before the test begins. The experiment is explained and practice is given in the different tests to be used. On the following Monday the squad reports at the scheduled hour, has initial observations made by the physiologist, dons a special uniform

consisting of underwear, khaki trousers and sweater, and enters the experiment room. The physiological observations include rectal temperature, blood pressure and pulse, reclining and standing, and respiration. This perhaps consumes twenty to thirty minutes. The psychologist then assumes charge of the squad and presents the various mental tests such as naming of 100 colored squares as quickly as possible, naming the opposite of words from a printed list of 50 words, cancelling certain digits from a sheet filled with numbers, adding columns of figures, multiplying a three digit by a three digit number mentally, the final answer only being recorded on the paper, copying on the typewriter certain extracts from a selected book. From 10 to 30 minutes is devoted to each of these tests. The physiological observations are then perhaps repeated at the middle of the period followed by a repetition of the mental tests and then a final observation by the physiologist. (See illustration, Fig. 19.) At the conclusion of the time the subject records his opinion as to his physical welfare in terms of the following scale:

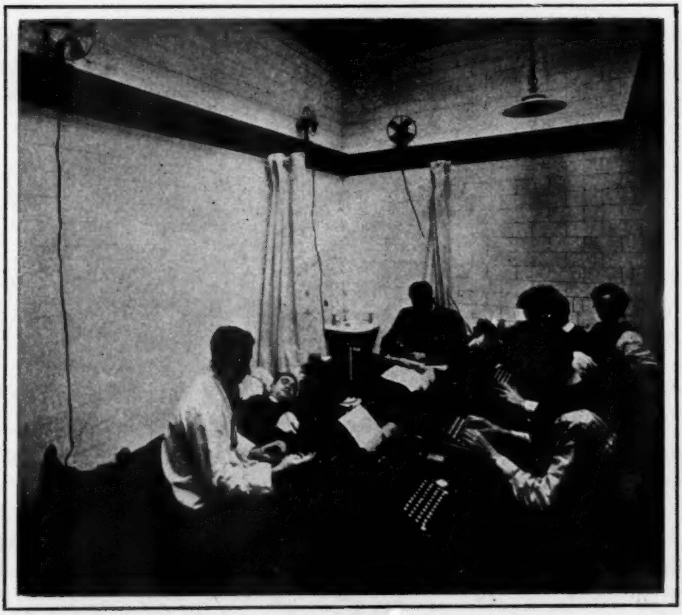


Fig. 19

Observation Room. The subjects are shown typewriting, one of the psychological tests used. The psychologist in the background is conducting the test. In the foreground the physiologist is shown taking a pulse reading on one of the subjects who reclines on a couch. The walls of this room were later covered with two inches of cork insulation to prevent heat loss.

5. I feel as comfortable as I ever do.
3. I feel about as I usually do at the close of an afternoon of hard mental work.
1. I feel as uncomfortable as I would with a severe headache or an attack of the grippe.
4. My condition is half-way between 3 and 5
2. My condition is half-way between 1 and 3.

SERIES I.

In the first series, lasting 11 weeks, the experimental period was from 2 to 6 p. m. five days a week, the conditions in the room being changed from day to day. A different squad of four men was used each week.

The average physical conditions in the room for the entire period are shown in Table 11.

TABLE 11

Series I

Room Conditions. Physiological Averages

11 Weeks—Days omitted when room conditions were not attained

Temperature			Air	CO_2 parts per 10,000			Number
D.B.	W.B.	R.H.	Supply*	2 P.M.	4 P.M.	6 P.M.	of days
86.5	80.	75.	45	6.0	5.5	6.0	19
87.0	79.5	72.	45 (electric fans)	5.0	8.5	8.5	7
68.5	57.	50.	45	6.5	...	7.0	18

* Cubic feet per minute per person.

The results of this work are illustrated in Figures 20, 21 and 22. The *temperature* of the body as registered in the rectum indicates a gradual rise during the 3½ hours in the 86 degree atmosphere. The electric fans playing in the faces of the subjects do not check this rise. At 68 degrees a rather marked drop of a half degree is noted. The final temperature at 86 degrees is about 37.51 degrees and at 68 degrees, 36.86 degrees, a difference of .65 degree.

Respiration is apparently not altered consistently by temperature, the breathing being slightly more rapid with the fans but showing little difference between 68 degrees and 86 degrees.

The *pulse rate* tends to decrease in the cooler room and the *blood pressure* to decrease less than in the warmer room. The relief afforded by the fans is not indicated.

The *Crampton value* is an arbitrary scale proposed by Dr. C. Ward Crampton as a measure of vasomotor tone. It expresses in per-

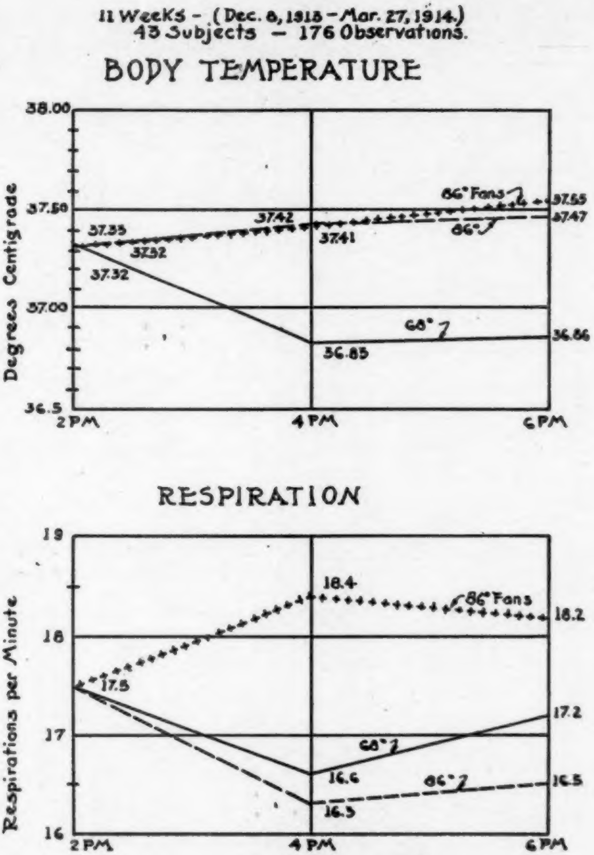


Fig. 20
SERIES I

centage the relation between change in heart rate and blood pressure on rising from a reclining to a standing position. One hundred per cent. on the Crampton scale means that the heart rate on standing increases less than 5 beats whereas the blood pressure increases by 10 mm. A zero value is represented by an increase of 40 beats in the heart rate on standing with a corresponding drop in blood pressure of more than 8 mm. Better tone or tension in the circulatory system with but a slight tax on the heart gives the higher value in the Crampton scale.

Expressed in these terms the 68 degrees temperature shows a marked improvement over the hot moist room.

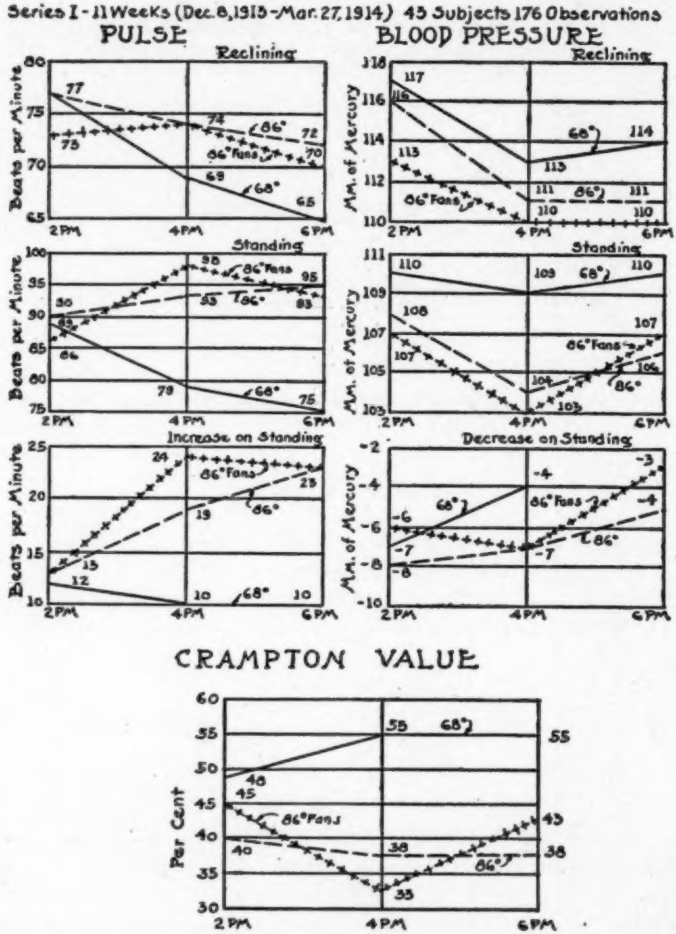


Fig. 21

The *mental tests* have been expressed on a per cent. basis, the 68 degrees condition serving as 100 per cent. The variation here is extremely slight and the variation is not consistent in the different tests.

The *comfort* of the subjects expressed in terms of the scale previously described favors the cooler room as would be expected. The relief of the fans is likewise indicated.

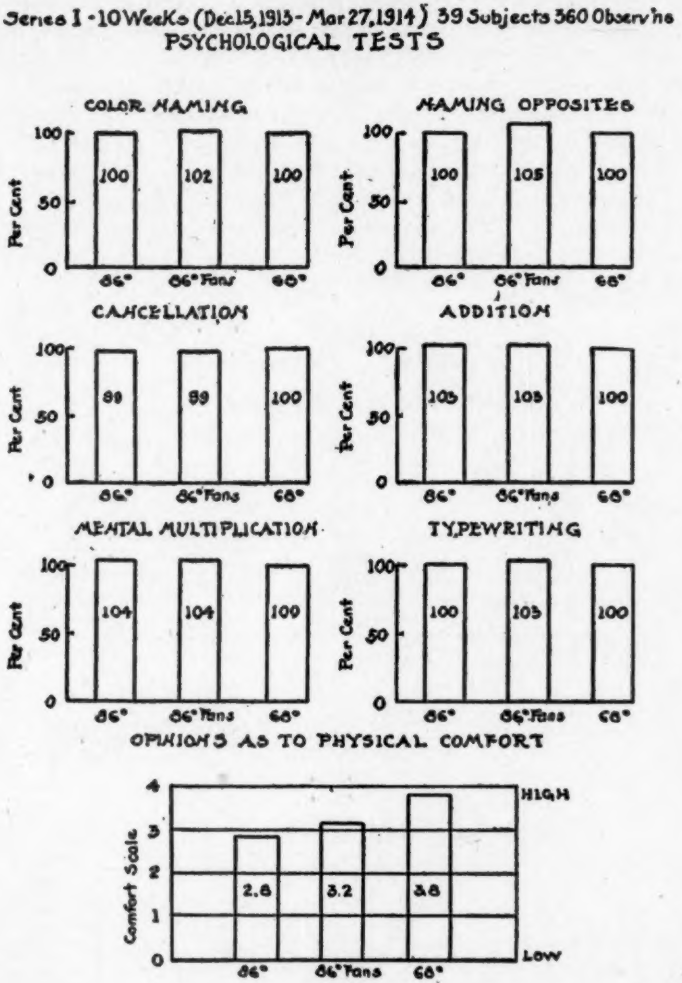


Fig. 22

SERIES II.

It was the intention in this series to study the accumulative effect of an air condition for five successive days. Three variables are introduced by reason of the failure to produce the hot stagnant condition until the sixth week, recirculated air being used in the first

two hot weeks. The physiological results are open to the above comparison but the continuity of the psychological tests permits of a division into only two conditions, hot and cool, four weeks of each. The squads were changed at the end of two weeks.

The experimental period was of the same duration as in the first series and the tests applied were very similar.

The average physical conditions in the room are shown in Table 12.

The results of the tests are shown in Figures 23, 24 and 25.

The same fall in *body temperature* is noted here in the 68 degree condition, the drop being 0.8 degrees, while the difference between the two hot conditions is barely perceptible.

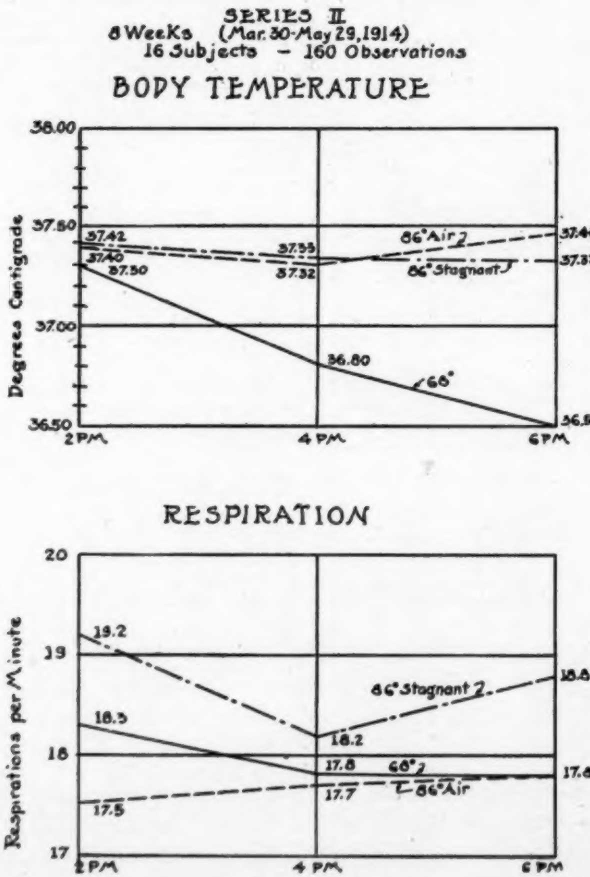


Fig. 23

TABLE 12

Series II

Room Conditions—Physiological Averages

Temperature D.B.	W.B.	R.H.	Air Supply*	CO_2 parts per 10,000 2 P.M.	6 P.M.	Number of days
86	80	77	40	5.0	7.0	12
86	81	80	0	6.0	32.0	8
68	58	52	45	4.0	5.5	20

* Cubic feet per person per minute.

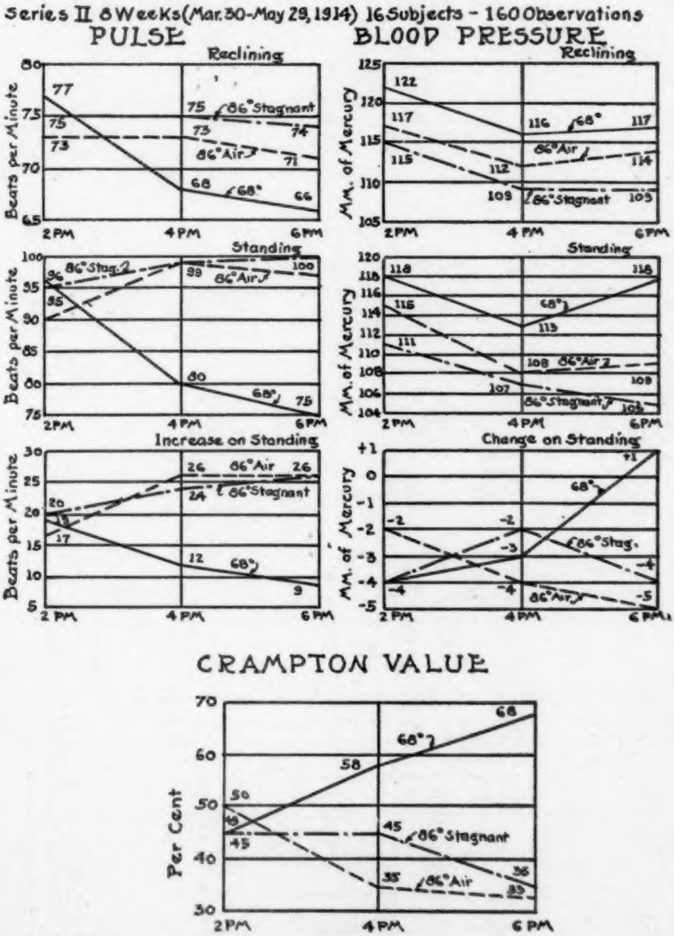


Fig. 24

No distinctive tendencies are noted in the *respiration* (Fig. 23) of the two hot conditions.

The days with air lie somewhat closer to the 68 degree condition but this is due to the positions of the initial *pulse* and *blood pressure* readings rather than to any change taking place during the experimental period.

The *Crampton value* again favors the cooler room.

Room temperature again fails to influence *mental efficiency* as shown in Fig. 25, although the feelings of the subjects differ materially as shown in the *comfort vote*.

The average room conditions for the psychological tests are shown in Table 13.

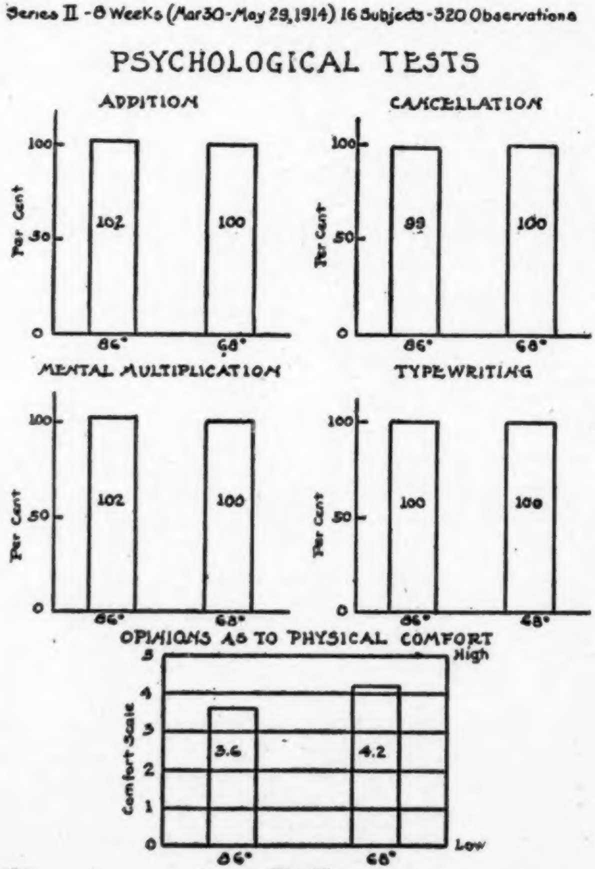


Fig. 25

TABLE 13

Series II

Room Conditions— Psychological Averages

Temperature D.B.	W.B.	R.H.	Air Supply*	CO_2 parts per 10,000 2 P.M.	4 P.M.	6 P.M.	Number of days
86 degrees	80	78	25	5.5	...	17.0	20
68 degrees	58	51	45	4.0	...	5.5	20

*Cubic feet per minute per person.

SERIES III.

In Series III, five successive days were given over to each air condition. Four men were retained throughout the six weeks. The period was lengthened to 8 hours, the subjects reporting at 8 a. m and remaining till about 4 p. m. Lunches were served in the Observation Room, the caloric value of the food consumed being noted.

The first three days of each week were spent in mental tests, the last two days being devoted largely to special work tests with the dumb-bell and on the bicycle ergometer.

The room conditions are shown in Table 14. The two weeks at 86 degrees with and without air supply have been grouped together as have the two weeks at 68 degrees. This arrangement accounts for the low average air supply and CO_2.

In Figures 26, 27, 28 and 29 are presented the results of the various observations.

TABLE 14

Series III

Room Condition—Physiological Averages

Temperature D.B.	W.B.	R.H.	Air Supply*	CO_2 parts per 10,000 9:00	11:30	2:00	3:30	Number of days
86	80	78	25	9.0	16.0	17.5	17.0	9
88	82.5	79	0 Fans	5.0	24.0	24.5	24.0	5
77	67.5	61	35	6.0	14.0	8.0	7.0	5
68	58.	55	25	9.5	....	26.5	27.0	10

* Cubic feet per person per minute.

Body temporature shows the influence of the environmental temperature as already brought out in the first two series whereas a difference in the amount of air supplied has no indicated effect whatever.

In Table 15 is given a summary of the room conditions for this air supply comparison.

TABLE 15

Series III

Room Conditions—Average for Air Supply Comparison

Temperature D.B.	Air Supply*	CO_2 parts per 10,000 9:00 A.M.	11:30	2:00 P.M.	3:30	Number of Observations
86, 77, 68 degrees	45	6.0	....	6.0	5.5	9
86, 77, 68 degrees	0	11.5	31.0	39.0	34.0	10

* Cubic feet per minute per person.

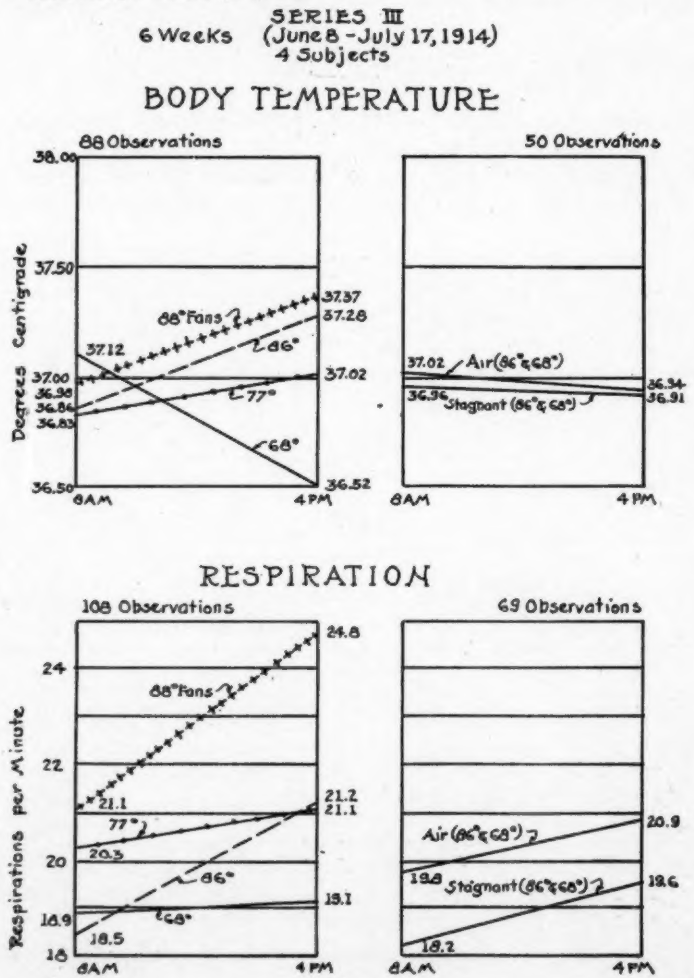


Fig. 26

Judging by the slope of the curve, *respiration* is increased by the warmer temperature, although the differences in the initial averages complicate this comparison somewhat. The increased breathing in the hct moist rooms with the fans was indicated in the first series. The chemical composition of the air apparently does not influence the rate of breathing.

In Fig. 27, the marked drop in *pulse rate* at 68 degrees is noticed. *Blood pressure* changes are not so consistent.

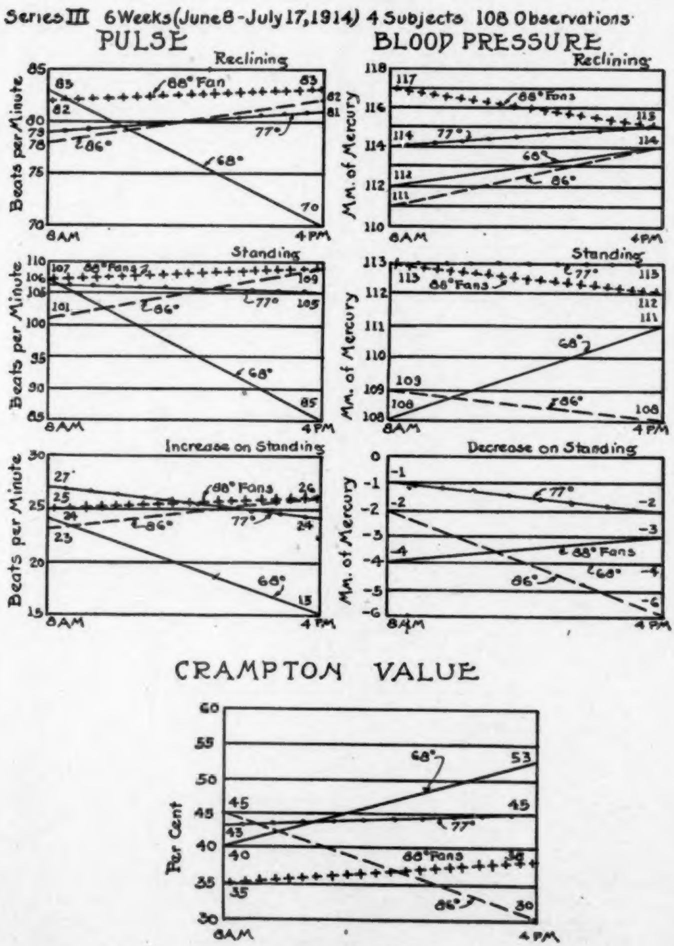


Fig. 27

In the *Crampton Scale* the room conditions range themselves in a regular order, 68 degrees showing to best advantage and 86 degrees the worst, the 77 degree and 88 degree fan condition occupying an intermediate position.

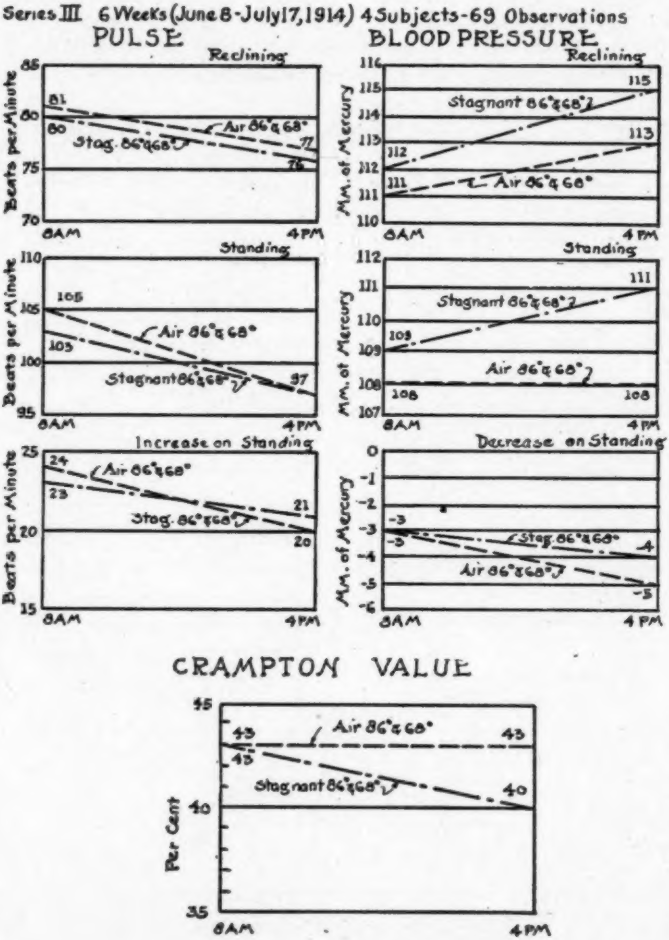


Fig. 28.

That pulse and blood pressure are unaffected by the chemical composition of the air is well illustrated in Fig. 28. These reactions are almost identical in a room with ample air supply and in a room with no fresh air supplied, the carbon dioxide accumulating to over 30 parts per 10,000. Temperature, however, has an entirely different effect, as shown in Fig. 27.

The effect of air conditions on *appetite* is illustrated in Fig. 29. The luncheon was altered each day but nearly the same type of food was served under each room condition so that the question of attractiveness of various meals is largely eliminated as a variable factor. Giving the smallest average calories consumed under any room condition a value of 100 per cent., the amount consumed in the other weeks is expressed as a positive value above 100 per cent.

More food is eaten at the lower temperature than at 86 degrees, but the increased consumption on the days with air supply is even

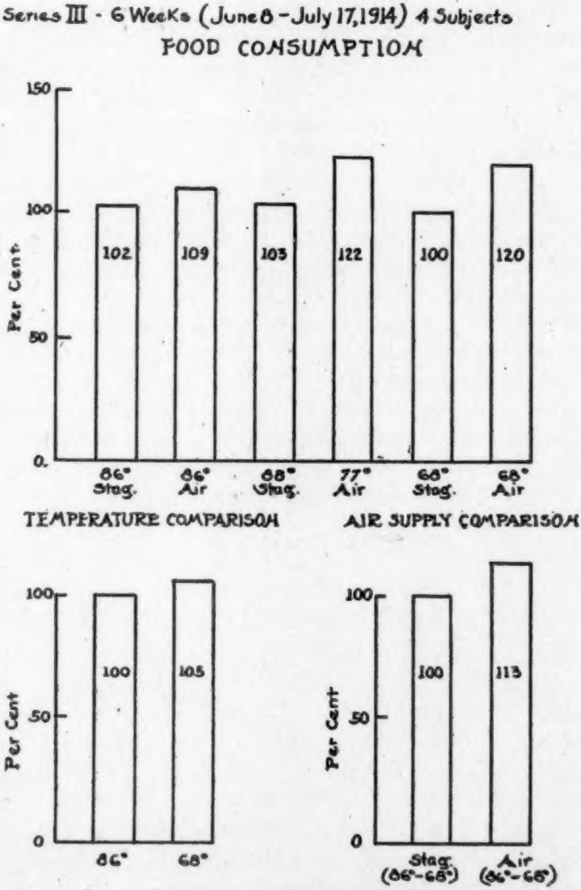


Fig. 29

more striking, 13 per cent. more food being eaten on the air days, at temperatures of 86 and 68 degrees.

The regularity of the *physical work* test was somewhat interrupted and the results from this are not as exact as the other tests. The physiologist in charge of the squad endeavored to keep at least one man and sometimes two at work constantly for about four hours in the two work days of each week. The men working in relays were urged on to their best efforts. A summary of these results indicated that 37 per cent. more was done at 68 degrees than at 86 degrees. That ample fresh air supply was not essential for physical work is suggested by the fact that 5 per cent. more work was done under the stagnant condition.

Three different *mental tests* were used, the purpose being to obtain a more delicate measure of mental efficiency than had hitherto been brought out in the other tests. Different specimens of handwriting were compared with a chart containing various samples of penmanship graded on a scale of excellence. The subject recorded his judgment as to the rank of the handwriting before him. These same specimens had previously been graded by a large number of qualified people the mean resulting score being used as a standard on which to rate the work of the subjects in our experiments. The result is expressed in terms of deviation from the mean value (see Table 16).

No falling off in judgment is indicated by exposure to the hot conditions, the slightly better score even favoring the warm days and the days with air supply.

The completion of couplets of poetry and the grading of short compositions were also used as tests in this series but the final scoring of the results has not yet been completed, owing to the excessive amount of work involved in framing a standard scale.

TABLE 16

Series III

Psychological Tests—Score in Grading Specimens of Handwriting

Room (D.B.)	Average Deviation from the Mean. (Low Value is the better score)	Number of Observations	Room (D.B.)	Air Supply	Average Deviation from the Mean	Number of days
		24	86 & 68 degrees	45	1.37	24
		12	86 & 68 degrees	0	1.43	23
		12				
		23				

SERIES IV.

Two subjects were selected for this series, lasting five weeks, during August and September. The entire time was devoted to intensive physiological studies, chiefly directed at changes in metabolism under

hot and cool conditions, with and without a supply of fresh air. In none of these tests, however, were consistent positive body responses shown that may be attributed to ventilation conditions. This work included studies on the duration of digestion, carbon dioxide content of the alveolar air, the oxyhæmoglobin content of the blood, heat production of the body, respiration rate, size of the dead space in the lungs, rate of pulse recovery after physical exertion, freezing point and specific gravity of the urine.

SERIES V.

The mental tests in Series I and II were efficiency tests, the subjects working at their maximum effort. Series V consisted of one week's work with four men, using option or inclination tests. The experimental period extended from 9 a. m. to 4 p. m. In addition to the regular wage the subjects were given a bonus for the number and correctness of mental multiplication problems done. A definite time in the morning and afternoon was set apart in which the subjects could either work on the multiplication examples for pay or else sit quietly and read novels. The option to the typewriting test was talking. A still further measure on their inclination was had by giving the option of reading a heavy scientific treatise or sleeping. Improvement from day to day in the first two tests was to be expected as the result of practice and the effect of any ventilation condition would only be shown by a perceptible dent or hump in the practice curves.

In Fig. 30 is shown the result of the daily scores along with the room conditions. Several points of interest are to be noted:

1. The falling off in mental multiplication on the fifth day, a hot, moist, stagnant day.
2. A perceptible improvement on the following cool day in mental multiplication, typewriting and heavy reading.
3. The fall in the typewriting curve on the fourth day.

This last point was due to the realization on the part of the subjects that more money could be earned in mental multiplication if they eased up on the typewriting test which just preceded it. This idea operated on the third and fourth days and entirely overshadows any variable that might have been caused as a result of ventilation conditions. That the idea was a fallacious one is shown by the improvement in both scores on the last day.

The men in this squad were also subjected to a rigid meat-free diet for eight days in the hopes of detecting the effects of hot, moist

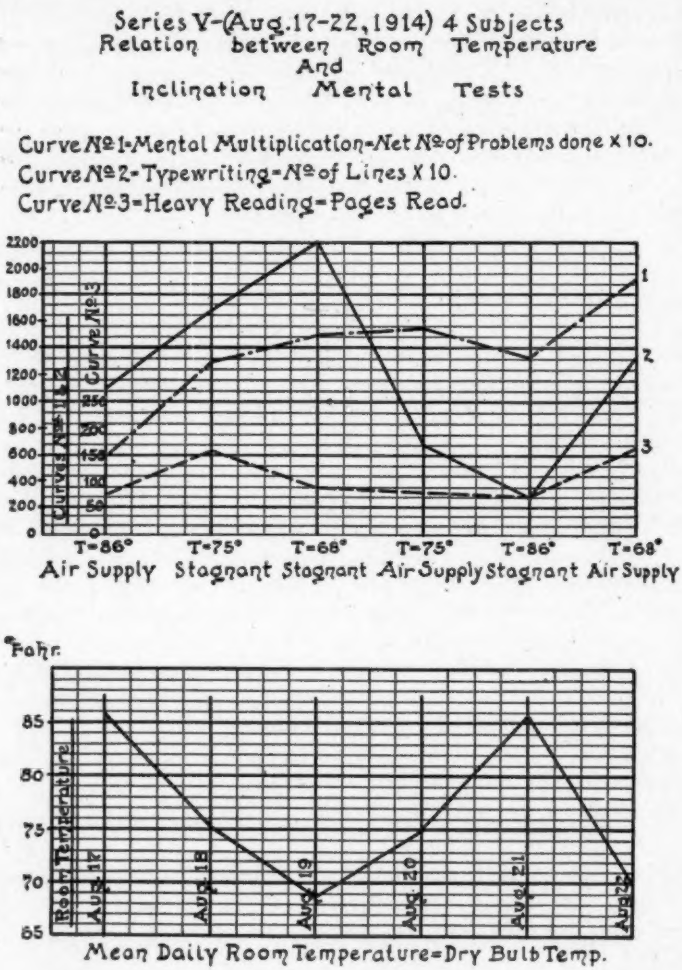


Fig. 30

air on the increase of creatinine in the urine. This substance would, under the circumstances, have to be recruited from the protein of the body tissue. Careful analysis failed to reveal any increase of this waste product on the hot days.

The humidity on the days at 86 degrees was 80 per cent., and on the other days between 50 and 60 per cent.

SERIES VI AND VIII.

The experiments to date had indicated that high temperature and in fact even 75 degrees affected, measurably, certain physiological reactions of the body but that mental processes were not impaired. The inclination test of the last series suggested a new method of approach. Series VI was thus designed to study inclination and maximum effort tests side by side at 68 and 75 degrees, the humidity remaining constant at 50 per cent. and the air supply omitted altogether in both cases. Five squads of 4 men each were used in this experiment, the room schedule being arranged as follows:

First week—68 degrees temperature all day.

Second week—68 degrees in the morning; 75 degrees in the afternoon.

Third week—75 degrees in the morning; 68 degrees in the afternoon.

Fourth week—First 3 days at 68 degrees; last 3 days at 75 degrees.

Fifth week—First 3 days at 75 degrees; last 3 days at 68 degrees.

The maximum effort addition test and mental multiplication and typewriting option tests were used.

In general the best accomplishment in the inclination tests took place in the cool period which occurred during the mornings in the first week and during the afternoons in the second week. This did not hold true, however, during the last two weeks where owing largely, we believe, to individual peculiarities in the squads, the final result favored the warmer temperature. In fact the warmer condition was even voted to be the more comfortable of the two.

In order to shed more light on this subject, this experiment was repeated as Series VIII, new groups of subjects being used. This second test did not entirely verify the result in the first.

Nearly 1,500 observations on 44 different individuals, during a period of nine weeks are thus summed up in the results of these two series. In Table 17 are presented the comparative scores on a per cent. basis for the half-day change and the 3 day change in the experiments as well as the grand average, including the two types of days in one grouping.

It would appear then from the above results that at the *same relative humidity* (1) the 75 degree condition is somewhat preferable for tasks involving deep concentration such as mental multiplication; (2) the 68 degree condition is slightly more desirable for combined mental and motor tasks such as typewriting; (3) that the difference between the two temperatures is practically negligible for maximum effort tests involving mental processes similar to those used in

additions of columns of figures; and (4) that there is no choice between these two variables so far as the physical comfort of the subject is concerned.

TABLE 17

Series VI and VIII

Comparison of Inclination and Maximal Effort Mental Tests at 68 and 75 Degree Air Temperatures
Per Cent.

Test	68 Degrees	75 degrees
Half Day Change		
Addition	100	97
Mental Multiplication Option	100	103
Novel Reading	100	79
Typewriting Option	100	75
Comfort Vote	4.0	4.1
Three Day Change		
Addition	100	100
Mental Multiplication Option	100	119
Novel Reading	100	84
Typewriting Option	100	107
Comfort Vote	4.3	4.2
Average of Half Day and Three Day Change		
Addition	100	99
Mental Multiplication Option	100	110
Novel Reading	100	81
Typewriting Option	100	94
Comfort Vote	4.1	4.1

SERIES VII.

The application of the inclination-to-do-work test to physical studies was instituted in this seventh series, to analyze further the relative effects of 68 degrees and 75 degrees and the importance of ample air supply. Four subjects were used to study.

(a) Accomplishment in voluntary physical work.

(b) Variations in appetite.

(c) Effect of extreme exertion on rate of pulse recovery.

(d) Effect on various physiological responses.

The experimental period was from 9 a. m. to 4 p. m. One hour was allowed morning and afternoon in which the subjects were privileged to earn extra pay by repeated lifting of a dumbbell a distance of about 2½ feet. Two hours were likewise set apart for maximum exertion on the bicycle ergometer with frequent observations of pulse before and after the exercise.

One hour at noon was given over to the lunch period. The remaining time was spent in rest and in making observations on body temperature, pulse and blood pressure.

The physical conditions of the room were changed daily and the resulting averages are shown in Table 18.

TABLE 18
Series VII
Room Conditions

Temperature			Air	CO_2 parts per 10,000				Number
D.B.	W.B.	R.H.	Supply*	9:00	11:30	2:00	3:30	of days
75	64	54	0	5.5	70.5	87.5	66.0	5
75	63	49	45	5.0	11.0	11.5	9.5	5
68	57	51	0	7.0	46.0	62.5	55.5	5
68	57	50	45	6.0	11.0	11.0	8.5	5
				Temperature Comparison				
75	63.5	52	25	5.5	41.0	49.5	38.0	10
68	57.0	50	25	6.5	28.5	37.0	32.0	10
				Air Supply Comparison				
75 & 68		52	0	6.5	58.0	75.0	61.0	10
75 & 68		50	45	5.5	11.0	11.0	9.0	10

* Cubic feet per minute per person.

The effect of the air variables on *body temperature* are shown in Table 19.

TABLE 19
Body Temperature

Room Condition	9 A.M.	2:45 P.M.	Change
75 degrees—Stagnant	37.04	36.94	— .10
75 degrees—Air	37.06	36.98	— .08
68 degrees—Stagnant	37.07	36.72	— .35
68 degrees—Air	36.91	36.73	— .18

Pulse and *blood pressure* showed scarcely any consistent differences. These results may be summed up briefly in Table 20, giving merely the Crampton values for the different conditions.

TABLE 20
Crampton Value (Per Cent.)

Room Condition	9:00 A. M.	2:45 P.M.
75 degrees—Stagnant	65	65
75 degrees—Air	68	63
68 degrees—Stagnant	70	65
68 degrees—Air	65	65

The result of the *work test* is shown in Fig. 31. Thus when left free to occupy their time, at either work or rest, the subjects performed 15 per cent. more work at 68 degrees than at 75 degrees. Two per cent. more work was done when air was supplied.

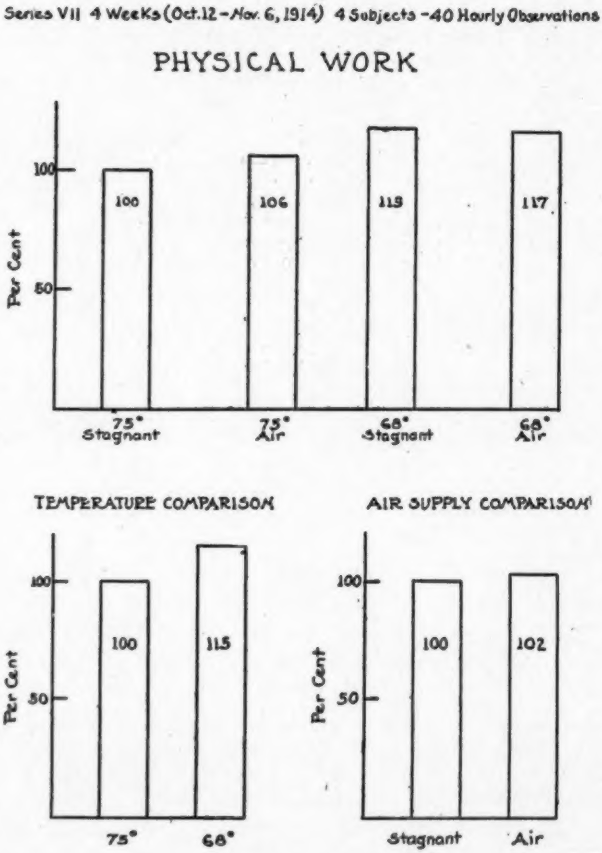


Fig. 31

The effect of these ventilation variables on the *appetite* is shown in Fig. 32. Whereas the cooler temperature favored an increase in physical work the appetite did not keep pace with this change. The largest food consumption occurred on the days with air supply.

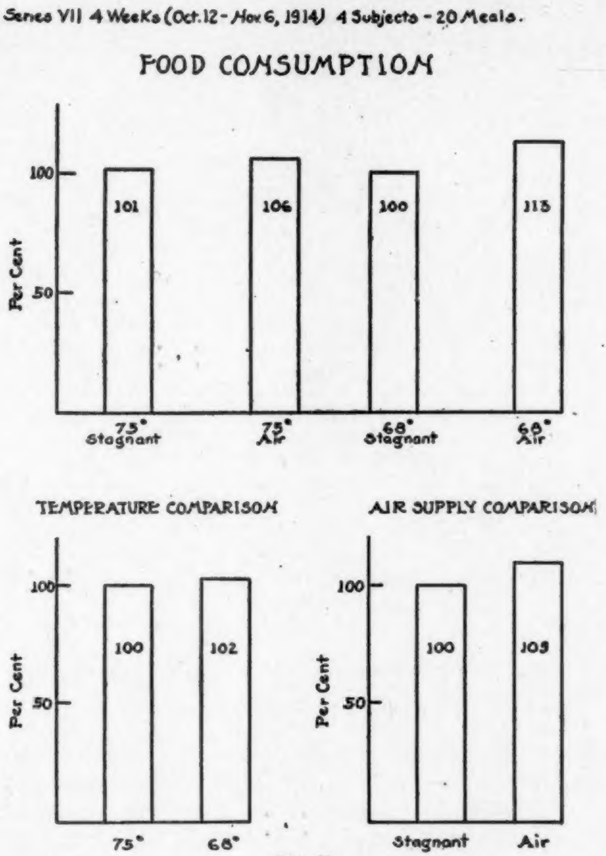


Fig. 32

The study on *pulse recovery* shows but slight difference due to air conditions. This difference, however, is consistent in indicating that the pulse tends to return somewhat more quickly to its initial rate when the temperature is 68 degrees than when it is 75 degrees (see Fig. 33).

No *comfort vote* was taken during this test. Casual observations by the person in charge of the squad were recorded, however, and these indicate no marked difference due to air supply. The warmer temperature is coincident with drowsiness and disinclination to work, whereas these factors are largely absent at 68 degrees. Playfulness and manifestation of surplus energy and tendency on the part of the

subjects to "rough-house" is recorded on several occasions but only on the cooler days.

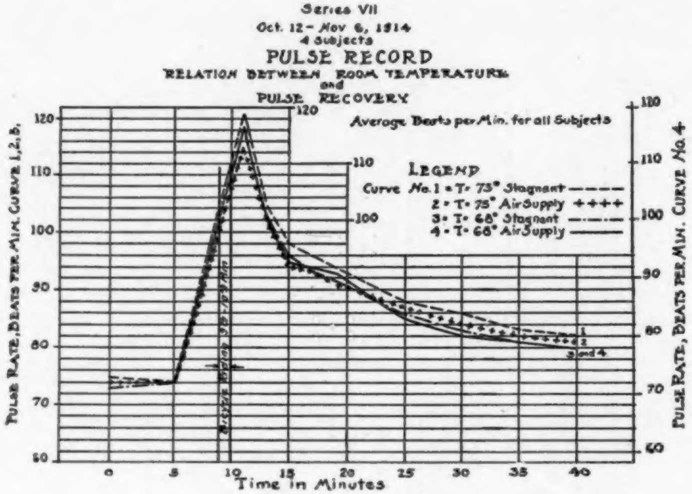


Fig. 33

GENERAL INFLUENCE OF ENVIRONMENTAL TEMPERATURE ON BODY TEMPERATURE.

Taking all the different series as a whole the most striking and consistent response to temperature changes in the surrounding air occurs in the temperature of the body. This correlation has been found to exist not only while the subjects have been in the experimental chamber but also while they are out of doors. In Fig. 34 is shown the relation of out-of-door temperature from 9 p. m. to 8 a. m. to body temperature taken between 8 and 9 a. m. The records have been expressed in terms of weekly averages.

SUMMARY.

It is difficult at this time to arrive at any sweeping conclusions as to the relative importance of different ventilation factors. Humidity has not been studied at all to date. The first year's work of the Commission has, however, developed the fact that:

1. Temperature within the range from 86 degrees to 68 degrees F. has a marked effect on certain physiological responses.

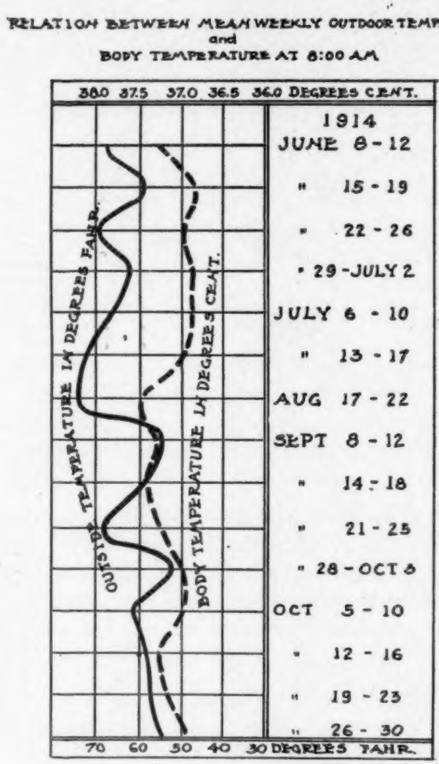


Fig. 34

2. Stagnant air, lacking a definite disagreeable odor, but containing all the products of the exhaled breath including carbon dioxide in excess of 30 parts per 10,000, is objectionable in a manner as yet unknown but demonstrated by a lessened desire for food, but otherwise shows no debilitating effect on the mental processes nor on the various physiological reactions which have been studied in these experiments.

DISCUSSION

Mr. D. D. Kimball: I am very sorry that the chief of our investigating staff, Mr. Geo. T. Palmer, could not be here this morning,

as the greatest amount of credit is due him for the preparation of this paper. As you will see it contains seventy-three pages which will take most of the morning session to read in full and I think you will be most interested in that part that relates to our experiments and results. I will, therefore, read only that which is necessary.

Mr. H. M. Hart: I want to ask a little explanation. See figure 11, if curve 2 was the theoretical carbon dioxide and curve 1, the actual carbon dioxide content. There is a falling off in the lower part of the table, between 3:30 and 4 p. m. How do you account for that?

Mr. D. D. Kimball: The students were leaving the rooms.

Mr. H. M. Hart: I think there is no doubt but what this low curve No. 1 was due to the leakage through small crevices and opening doors and windows, because in our experiments at Chicago we had the carbon dioxide so high that we could not measure it with the ordinary machine. It was over 250.

Prof. C. E. A. Winslow: Mr. Chairman and Gentlemen, I wish to speak a word of caution as to the interpretation of these physiological results. The differences as indicated in table 17 of the report are very slight really for work of that character and the psychologist to the Commission don't consider these differences significant.

I think that the net result of the work as far as these maximum efficiency tests are concerned is negligible and probably also negligible as far as the option tests go. I think this is due in part to the insensitiveness of the subjects to these conditions. I don't think the results show anything positive in regard to the effect of air conditions, neither do I think that it can be considered as a negligible result applicable to all people. We are planning to repeat these experiments with women who, I suspect, will be found more sensitive to heat than the young men.

As to the interpretation of these results, they compare with other physiological experiments in the past and are very conservative. On the other hand, they strike the public as radical and some newspaper editors got in their deadly work when they were presented.

The results do not in any sense indicate that ventilation is not necessary. On the contrary, they furnish very excellent evidence that ventilation is necessary for the removal of heat from the body. The tendency is simply to show that the discomfort in badly ven-

tilated rooms is due mainly to temperature and humidity. I think we can claim that, even if the appetite experiments are not confirmed by our subsequent work. If they are, they furnish the first clear evidence that there is something more than heat and humidity. The public decency is involved in all these questions as well as the public health.

Even if it should be shown that cool, stagnant air is not harmful to health, it is entirely proper to insist that rooms in schools and where people congregate, should not smell bad. It is the same argument that would apply to the personal bathing of the body.

Ventilation is necessary to remove odors as well as to keep down the temperature, even if the odors do not affect the health. The results, I think, give us stronger evidence than we have had before as to the practical harm of overheating.

The appetite experiments are particularly new and if future experiments substantiate the first two series, it is possible that we shall have much clearer evidence than ever before as to the serious effects of bad air.

I wish to say one word about the personnel of the Commission. We have a well balanced commission of one medical man, and one engineer, the other members of the Commission being a physiologist, chemist, psychologist and a sanitarian, so I think it might be said that this Commission represents all classes of workers interested in the subject and have been from the first during its year and a half's work, entirely unanimous in the experiments and interpretation of the results.

In the final report we shall be able to produce results which will be unanimous and express clearly the views of all the groups of workers concerned.

Mr. J. H. Davis: I am very glad that Prof. Winslow made this explanation, because he has certainly been misquoted in various trade papers and certain societies, that the New York Commission have decided that ventilation is practically unnecessary. I have the privilege of being a member of the Chicago Commission on ventilation and we have found that this quotation has caused us some trouble and the explanation as made here that ventilation is not only desirable but necessary, will go a long way towards removing that impression.

Pres. Lewis: It would be highly desirable if Prof. Winslow's remarks could be given the very widest publicity. There has been a feeling in Chicago that we are not in harmony with the New York Commission.

Mr. F. K. Davis: Has any effort been made to get the previous history of the personnel of the subject used in the experiments? I find a wide variation in the individual as to his requirements. Some people desire a great deal higher temperature and others a lower. That occurred to me as something that might affect the experiments that were made.

Mr. D. D. Kimball: The subjects used were all City College boys and they were questioned as to their general condition and general health determined as far as possible and record of their general conduct, doings and actions were kept as far as possible. In the experiments with women we will probably go further. So far as possible, any facts coming to us that would affect our results were carefully noted and allowed for.

Pres. Lewis: Do you wish to say anything so that it may be entirely clear to all members, of the necessity of experiments with a large number of subjects so as to remove the possibility of the personal element appearing in the results? Is there anything to be said on that subject to make it clearer?

Mr. D. D. Kimball: Some of the earlier tests were for periods of four hours, and some of the experiments were carried on for five days in the week.

Pres. Lewis: We might have a body of ten or fifteen men who showed a marked decrease in experiments lasting one week and that might not mean anything. It may require a longer period to determine the actual effect of the experiment on a subject.

CCCLXIX

STUDIES IN AIR CLEANLINESS

GEORGE C. WHIPPLE* AND MELVILLE C. WHIPPLE†

Wherever man establishes his permanent abode he is confronted with the necessity of exercising in some degree the virtue of cleanliness. His very existence may depend upon this activity. The lone settler in a wilderness does not concern himself with conditions beyond his own door yard, the forces of Nature maintain the fitness of his environment; but when people elect to live closely together in towns and cities the problems that confront their pursuit of health and happiness become numerous and varied. The occupations and activities of a large population are accompanied by many processes of waste and destruction, that left uncontrolled inflict hardships upon his life and community.

The necessity for maintaining a pure water supply and an adequate system for the disposal of refuse and sewage, the desirability of clean streets, and the benefits that arise from well ordered recreation centers, are all recognized as being essential to the public welfare. The importance of a clean air supply has not been so firmly established in the mind of the average citizen. This may have been due to the lack of certain fundamental information upon the qualities of air and its physiological relations, and to a belief bred of such ignorance that certain evils were necessary ones. Some of these things have now come to be much better understood and considerable attention is being given to a study of the whole problem. It is the purpose of this paper to present the results of some studies in the quality of city air as regards its suspended impurities and the influences that effect the distribution of these impurities.

During the past two years the authors have conducted numerous investigations bearing upon the dust and bacteria content of air at different places in various cities. For instance, a study of the air above the streets, made from high buildings in Boston and New

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York, showed a constant decrease in the number of dust particles with increasing elevation. At the fifth story the number averaged slightly less than half the number at the street level, and at a height of 600 to 700 feet the air approached closely in purity that over Long Island Sound. The air in suburban districts was found to contain less dust and fewer bacteria than that in the centers of activity. Observations made after hard rains showed that there was a very effectual removal of suspended impurities at such times. High winds, on the other hand, were found to increase the dust and bacteria. The facts of many of these findings were already known in a general way, but the results that were obtained established definite measures of difference.

In the course of the investigations the process of air washing was given considerable study. This is a means for purification of unclean air that is destined to receive a great deal of attention in satisfying a more widespread demand for clean air in our buildings. Properly controlled this process is capable of furnishing air containing very little dust, few bacteria and of any desired temperature and humidity. It assures a supply of air of uniformly good quality. This is not possible if untreated air is taken into city buildings, particularly if they are located in congested or dirty districts.

Several series of experiments were made in Cambridge, Mass. during the past year for the purpose of studying the distribution of atmospheric dust in different districts of the city. The work was done at the instance of the Cambridge Sanitary Commission. The previous method of intermittent sampling of the air under different conditions was abandoned for one that registered the total precipitation of dust on a given surface over a period of several days. The advantage of the latter method is that it takes account of conditions during the twenty-four hours, and during any changes that occur from day to day. Any attempt to cover the same range by a count of dust particles in individual samples of air would involve an immense expenditure of time and effort.

For collection of the dust two-quart tin pails were used. These were coated with resistant varnish inside and out, about one liter of distilled water was placed in them, and they were suspended from the brackets on the poles of the Cambridge Electric Light Co. The height above the street varied from 20 to 21 feet. Later, when the varnish showed signs of cracking from exposure to air and water, glass jars were inserted in the pails to avoid any sources of error from the varnish or from corrosion of the pails.

The water served to retain any particles that settled out on its surface. About two weeks' exposure was allowed for each series of experiments and in this time evaporation did not exceed the liter of water originally placed in the pail. It averaged something less than 500 c.c. When the samples were brought into the laboratory the contents of the pails were carefully rinsed and each made up to one liter. Analyses were made of this water and the analyses when expressed in parts per million represented milligrams per liter, that is total milligrams actually collected.

Various determinations were made at the beginning of the work to obtain an idea of the amount of material collected, and it was found that the total solids obtained by evaporation of the water best represented this quantity, and it was considered that they supplied the most correct measure of suspended atmospheric impurities. Results for suspended solids in the water were not found to represent the suspended impurities on account of the solubility of portions of the atmospheric dust in the water in the pails. Filtration of the suspended solids showed effects of leaching, such as loss of color, and the filtrate was found to contain considerable traces of calcium, magnesium and chlorides.

Microscopical examinations were also made of a large number of samples but it was impossible to establish characteristic differences between them. The amount of agitation given the sample seriously affected the size of the particles which tended to agglomerate while in the pail. It was noticed by microscopical examination that the number of molds increased greatly in the samples taken in the summer over those taken in the early spring.

Turbidity readings were taken on all samples and in the later work determinations were made of silica and iron.

For purposes of study all samples have been divided into three series, according to the particular time at which they were collected. Series I represents the period from April 2-14 inclusive, Series II, June 29-July 13, and Series III, July 31-August 14, 1914. All the arrangements for samples and analyses in connection with Series II and III were made by Mr. L. T. Fairhall of the Chemical Department at Harvard University.

In Table I is given a summary of weather conditions for the three periods. The first period was windy, the velocity reaching a high figure on two or three days. The winter's accumulation of street dirt was also being removed at this time. Series II and III were taken during normal summer weather, except for a considerable amount of rain during Series II.

TABLE I

Table Showing Amount of Rainfall and Wind Movement during Periods when Dust was Collected from the Air, Cambridge, Mass.

(Compiled from Records of Boston Office of U. S. Weather Bureau)

Series Number	Date 1914	Rainfall in Inches	No. Days Rain Fell	Total Miles Wind Movement	Maximum Wind Velocity, Miles per Hour
I	April 2-14	1.67	5	3469	47
II	June 29-July 13........	3.00	7	3091	25
III	July 31-August 14.....	0.45	4	3195	22

The results of all total solids determinations have been arranged in order of magnitude and by series in Table II. The mean for each series is given, and also the calculated amount of solids in milligrams that would have fallen in one day upon a surface one square meter in area, and again in pounds per acre per day.

TABLE II

Table Showing Total Solids Collected at Different Times from Air in Cambridge, Mass.

Series Number I. Date, April 2-14, 1914.

Location of Sample	Total Solids in Milligrams	Total Solids. Mg/Sq Meter/Day	Total Solids. Lbs./Acre/Day
Brattle St. and Sparks St...........	184	1063	9.5
DeWolf St. and Cowperthwait St...	229	1322	11.8
Mt. Auburn St. and Hawthorne St...	252	1458	13.0
Garden St. and Concord Ave........	255	1472	13.1
Shepard St. and Garden St...........	290	1680	14.9
Broadway and Dana St.	292	1690	15.0
Shepard St. and Walker St.	293	1696	15.1
Mass. Ave. and Waterhouse St.	303	1753	15.6
Brattle St. and Ash St.	332	1920	17.1
Harvard St. and Dana St...........	347	2005	17.8
Boylston St. at Power House........	370	2141	19.0
Kirkland St. and Irving St..........	374	2163	19.2
Cambridge St. and Kirkland St.....	421	2430	21.6
Kirkland St. and Oxford St.	430	2483	22.0
Mass. Ave. and Arlington St.	454	2622	23.2
Oxford St. at Pierce Hall...........	497	2876	25.6
Putnam Ave. and Franklin St.	534	3085	27.4
Mass. Ave. and Shepard St.	606	3510	31.2
Cambridge St. and Ellery St.	638	3690	32.8
Mass. Ave. East of Quincy Sq.......	695	4020	35.8
Huron Ave. and Concord Ave.	951	5500	48.9
Mean	417	2400	21.4
Median	370	2141	19.0

TABLE II (Continued)

Table Showing Total Solids Collected at Different Times from Air in Cambridge, Mass.

Series Number II. Date, June 29 to July 13, 1914

Location of Sample	Total Solids in Milligrams	Total Solids in Mg/Sq. Meter/Day	Total Solids. Lbs./Acre/Day
Brattle St. and Fayerweather St.....	102	506	4.5
Mt. Auburn St. and Maynard Pl. ...	125	621	5.5
Brattle St. and Ash St.	126	625	5.6
Brattle St. and Craigie St.	142	705	6.3
Brattle St. and Fresh Pond P'kway..	150	744	6.6
Elmwood Ave. and Fresh Pond P'k\	150	744	6.6
Cambridge St. and Maple Ave.......	165	819	7.3
Hampshire St. and Elm St.	174	863	7.7
Hampshire St. and Plymouth St. ...	177	878	7.8
Hampshire St. and Amory St.	182	903	8.0
Hampshire St. and Tremont St.	299	1483	13.2
Mechanics Square	892	4425	39.4
Mean	224	1110	9.9
Median	158	782	7.0

Series Number III. Date, July 31 to August 14, 1914.

Location of Sample	Total Solids in Milligrams	Total Solids in Mg/Sq. Meter/Day	Total Solids. Lbs./Acre/Day
Arlington St. and Washington Ave...	47	297	2.6
Linnaean St. and Raymond St.	56	354	3.1
Magazine St. and Erie St.	61	386	3.4
Brattle St. and Hawthorne St.......	73	462	4.1
Holmes Pl. and Peabody St.	76	481	4.3
Quincy St. opp. Colonial Club	78	493	4.4
Garden St. and Appian Way	82	518	4.6
Mt. Auburn St. and Elmwood Ave. ..	93	587	5.2
Beaver St. and Cowperthwait St.	95	600	5.3
Mt. Auburn St. and Aberdeen Ave...	97	613	5.4
Broadway and Ellsworth Ave.	98	619	5.5
Cambridge St. and Quincy St.	99	626	5.6
Mass. Ave. and Martin St.	141	892	7.9
Regent St. near R. R.	186	1175	10.5
Western Ave. and Hews St.	196	1238	11.0
Hampshire St. and Elm St.	220	1390	12.4
Kendall Square	332	2098	18.7
Mass. Ave. North of M. I. T. Bldgs..	380	2400	21.4
Mechanics Square	414	2618	23.2
Mean	149	939	8.3
Median	97	613	5.4

The extremes in each series were widely separated in quantity. In Series I the maximum amount of solids collected was about five times that of the minimum amount, in Series II and III it is about nine times. An attempt was made to have the sampling stations scattered in all classes of districts throughout the city from the residential to the manufacturing sections. Series I had no samples in purely industrial districts. It represented residence and

business districts, and included local ways and main thoroughfares. A more general representation was included in Series II and III. The locations at which the smallest amount of solids was collected were in the residence districts in each set of experiments, while the largest amounts were collected near the industrial centers or in portions of the city where conditions favored dust formation. The effect of an industrial center with large factories and poor streets upon the atmosphere of the surrounding district is shown in Series II and III. The highest figures for total solids were obtained at Mechanics Square. This is the center of a very poor district of the city in which there are numerous factories and plants, and streets in poor condition. The exceptionally high figure in the second series might easily have been due to a cleansing of the air from small particles of dust and soot during the rainfall of that period. There were two or three hard showers.

Of interest in connection with the distribution of dust over the city was a spot map prepared from the results of these experiments. A median figure was taken for each set of experiments. Upon a large map of the city pins were placed at the location of each station, the pin being green if the amount of dust collected at that station was less than the median figure, and red if it was more than the median. A survey of all the streets in the city had been made during the period of dust collection and a rating of good, fair, bad and indifferent given to them in accordance with their state of repair and cleanliness. Upon the map the rating was indicated by coloring the street, or portion of it blue, green, red or yellow respectively to conform with the rating given. After all the pins had been placed it was observed that practically all the green pins were in the blue or green street areas, and the red pins in the red or yellow street areas. In other words, the low dust figures were obtained on streets where the roadway was in good condition, well oiled, or clean, and the high dust figures on streets that were rated as in poor condition or dirty.

The exception to this distribution was in the case of the main thoroughfares that were rated in good condition and that had street car lines upon them. Red pins, representing high figures for dust were found upon several of these. Massachusetts Avenue, the main artery through the city, had figures above the median throughout its length as did Cambridge Street. Mt. Auburn Street, another through street with a car line and good pavement, had low figures for dust. This was explained by the fact that Mt. Auburn Street borders a parkway along the Charles River,

and has very few buildings on that side of the street and almost no cross traffic along the portion of it where sampling stations were located.

Comparison of the results obtained in each of the three series of experiments shows that considerably more dust was collected for the period between April 2 and 14 than for the other two. The mean of all the results for this period showed that solids were collected at the rate of 2,409 milligrams per square meter per day; for the period between June 24 and July 13 the amount was 1,110 milligrams; and between July 31 and August 14, 939 milligrams. The differences are consistent with the conditions prevailing during each series. In the first a large proportion of the winter's accumulation of dirt laid on the streets or was in process of removal. The total wind movement on each of three days was in excess of 300 miles and on April 12 reached a velocity of 47 miles per hour. None of the streets had been oiled at this time, a factor which no doubt exerted considerable influence as there is a very large mileage of streets in Cambridge that have dirt or macadam surfaces.

The wind movement during the second and third series was about the same for each, but on three days of the second series there was rainfall in excess of 0.6 inches. These showers were five days apart with very little precipitation on the days between them. Inasmuch as heavy rains have a cleansing effect on the air it is probable that considerable amounts of fine particles were precipitated by these showers.

In Diagrams 1, 2 and 3 the determinations of each series have been arranged in order of magnitude and divided into two groups, according to whether the sample was taken on a street with a car line or without. In Diagram I. a division is also made on the basis of paved and unpaved streets. The heavy center line at the intersection of the cross-hatching divides the two groups in each case. On account of the undue weight given to certain extreme results in computing the mean or average results the median figure as well as the mean has been indicated on the diagrams. A key to the location of samples is given in Tables IV-VI.

It will be noticed that in each diagram under the head of total solids both the mean and median figures are higher for those samples collected on streets with car lines than for those collected on streets with no car lines. This is not on account of the poor condition of the pavements on car line streets, for with only one exception the survey showed the surface of such streets to be

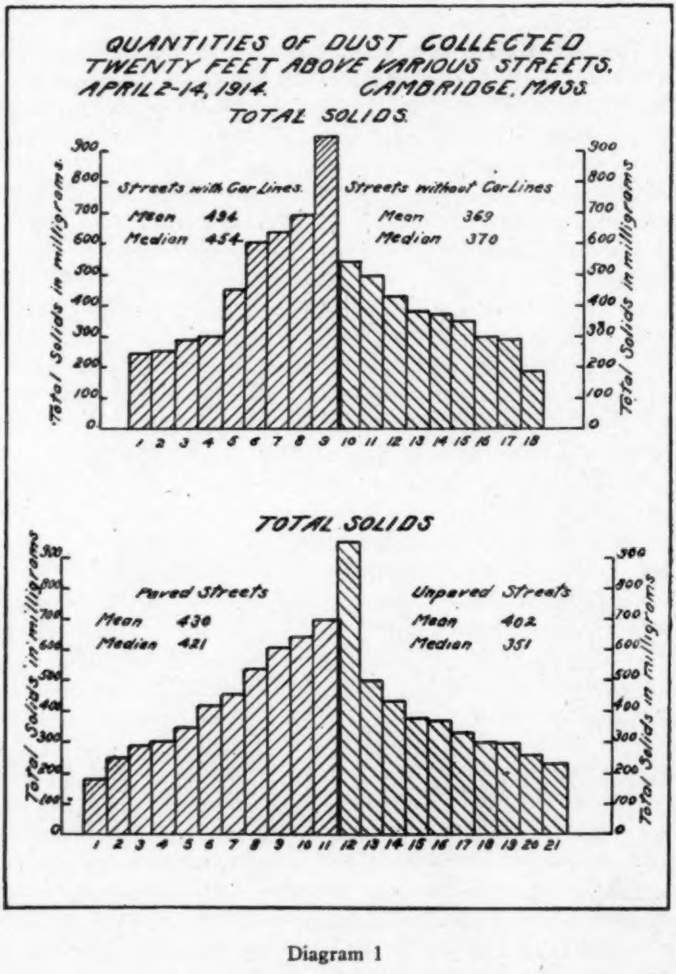

QUANTITIES OF DUST COLLECTED
TWENTY FEET ABOVE VARIOUS STREETS.
APRIL 2-14, 1914. CAMBRIDGE, MASS.
TOTAL SOLIDS.
Streets with Car Lines.
Mean 494
Median 454
Streets without Car Lines
Mean 369
Median 370
Total Solids in milligrams.
TOTAL SOLIDS
Paved Streets
Mean 430
Median 421
Unpaved Streets
Mean 402
Median 351
Total Solids in milligrams

Diagram 1

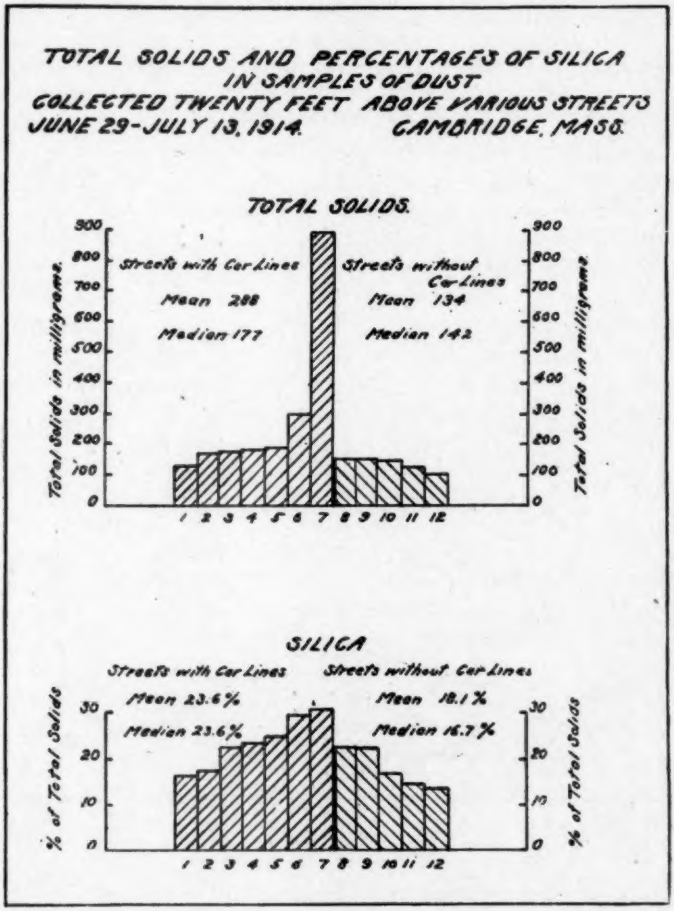

TOTAL SOLIDS AND PERCENTAGES OF SILICA
IN SAMPLES OF DUST
COLLECTED TWENTY FEET ABOVE VARIOUS STREETS
JUNE 29-JULY 13, 1914.
CAMBRIDGE, MASS.
TOTAL SOLIDS.
Streets with Car Lines
Mean 288
Median 177
Streets without Car Lines
Mean 134
Median 142
Total Solids in milligrams
0
100
200
300
400
500
600
700
800
900
1 2 3 4 5 6 7 8 9 10 11 12
SILICA
Streets with Car Lines
Mean 23.6%
Median 23.6%
Streets without Car Lines
Mean 18.1%
Median 16.7%
% of Total Solids
0
10
20
30
1 2 3 4 5 6 7 8 9 10 11 12

Diagram 2

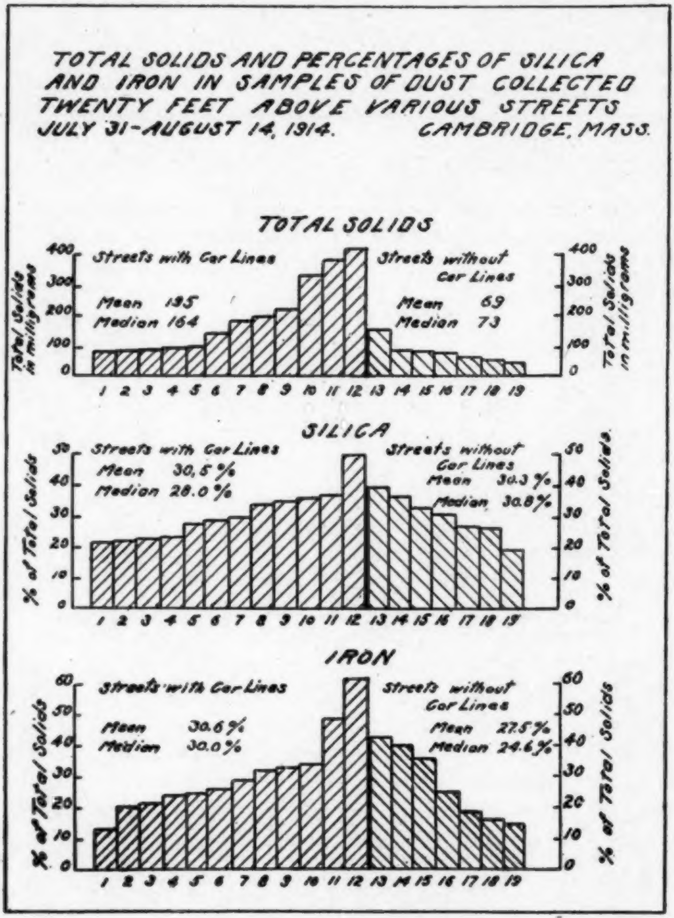


Diagram 3

in a good state of repair and throughout the greater part of their length these streets were covered with a smooth surfaced material, such as asphalt, bitulithic or wood block pavement. The controlling factor in such cases is the rapid motion of car traffic over the streets. Unless the pavement between and close to the rails is kept clean a cloud of dust is raised with the passage of each car. The height and extent to which this cloud is scattered can be seen very well at night near street lights where a dark ground

illumination occurs. The material scattered about consists of organic debris, gritty siliceous material and metallic dust from rails, wheels and brake shoes. All streets over which surface cars are operated should receive special attention from the street cleaning department. If they do not the streets not only lose in attractiveness for business and residence purposes, but they give rise to a menace to public health in the dust and infection that is scattered broadcast over them.

Diagram I. also shows the relative amounts of dust deposited over paved and unpaved streets. At the period when the samples were taken the unpaved streets were in their worst condition, as they had not yet been oiled. Inasmuch as the paved streets showed larger amounts of dust than the unpaved it is assumed that the effect of car traffic is reflected again in this diagram. Nearly all the paved streets had car lines running over them.

Full knowledge of the significance of dust in the atmosphere and its effects upon the health of the individual is not possessed by scientists at the present time. So far as these things have been studied, principally in connection with dusty trades, the opinion has been formed that the most dangerous dust particles are those of a gritty, rough nature. There is a large amount of such material in street dust.

As the studies outlined in this paper progressed the idea was suggested that a determination of the siliceous material and iron in the solids collected might be of interest. Accordingly, analyses of silica and iron were made on some of the samples. The results are presented graphically in Diagrams 2 and 3 where the amounts of these substances are shown in percentages of the total solids collected, and in Table III where the average figures for the amounts actually collected are given.

TABLE III

Average Results of Solids Determinations Expressed in Milligrams of Material Collected

	Total Solids		Organic Solids		Silica		Iron	
	Cars	No Cars	Cars	No Cars	Cars	No Cars	Cars	No Cars
					Series I			
Mean........	494	369	86	75	..	..	..	..
Median......	454	370	88	68	..	..	..	..
					Series II			
Mean........	288	134	135	74	55	25	..	..
Median......	177	142	86	73	41	21	..	..
					Series III			
Mean........	195	69	63	37	64	22	56	18
Median......	165	73	60	38	47	24	43	18

The results are grouped in the diagrams according to the streets that had, or did not have, surface car lines. The percentage differences are not great, but in each case the average results are somewhat higher for paved streets. The figures do not necessarily imply that the same percentages of silica and iron existed in the street dust as in the dust of the air. It was undoubtedly higher in the former, for silica and iron have a higher specific gravity than organic matter and would tend to settle out of the air more quickly than the latter. As already stated the samples were collected twenty feet above the street.

Drawing our conclusions strictly from these particular experiments and without attempting to apply them universally, it may be said that the popular belief is well founded, so far as Cambridge is concerned, that in the industrial communities of a city and in those sections where little attention is given to the conditions of the streets the air suffers a loss of cleanliness. It has been actually demonstrated that the amount of precipitated impurities is greater in such places than in the better groomed districts. It has also been shown that the nature and volume of the traffic on streets influences the cleanliness of the air above them. Streets with smooth and fairly clean pavements have afforded examples of a greater precipitation of dust than other streets the surface of which was not so well prepared nor so well cleaned.

The survey of atmospheric conditions in cities is a field of investigation that is rather new to sanitary science, but one that will be thought of seriously as our interest in such matters is quickened by an increasing knowledge of subjects that relate to the air we breathe. It is the opinion of the authors that the studies in Cambridge, as outlined in this paper, suggest a method of undertaking such surveys, which with refinements and adjustment to particular conditions will prove valuable in the information it furnishes. By its use an idea may be gained of the total amount and character of the solids precipitated from the air in a given period of days, the distribution of these suspended impurities geographically, and the extent to which their quantity is affected by varying weather conditions. Interpretation of the results, in connection with a thorough knowledge of the streets, life and industries in the districts investigated, will generally suggest specific remedies for the improvement of these districts.

The above investigation was made in the Laboratory of Sanitary Engineering of Harvard University and was made possible by funds kindly furnished by Mr. Ernest C. Dane of Brookline.

TABLE IV

Key to Location of Samples, Analyses of which appear in Diagram 1

Determination	Sample Number	Location
Total Solids on Streets with Car Lines.	1	Mt. Auburn St. and Hawthorne St.
	2	Garden St. and Concord Ave.
	3	Broadway and Dana St.
	4	Massachusetts Ave. and Waterhouse St.
	5	Massachusetts Ave. and Arlington St.
	6	Massachusetts Ave. and Shepard St.
	7	Cambridge St. and Ellery St.
	8	Massachusetts Ave. East of Quincy Sq.
	9	Huron Ave. and Concord Ave.
Total Solids on Streets without Car Lines.	10	Putnam Ave. and Franklin St.
	11	Oxford St. at Pierce Hall.
	12	Kirkland St. and Oxford St.
	13	Kirkland St. and Irving St.
	14	Boylston St. at Power House.
	15	Harvard St. and Dana St.
	16	Shepard St. and Walker St.
	17	Shepard St. and Garden St.
	18	Brattle St. and Sparks St.
Total Solids on Streets with Pavements.	1	Brattle St. and Sparks St.
	2	Mt. Auburn St. and Hawthorne.
	3	Broadway and Dana St.
	4	Massachusetts Ave. and Waterhouse St.
	5	Harvard St. and Dana St.
	6	Cambridge St. and Kirkland St.
	7	Massachusetts Ave. and Arlington St.
	8	Putnam Ave. and Franklin St.
	9	Massachusetts Ave. and Shepard St.
	10	Cambridge St. and Ellery St.
	11	Massachusetts Ave. East of Quincy Sq.
Total Solids on Streets without Pavements.	12	Huron Ave. and Concord Ave.
	13	Oxford St. at Pierce Hall.
	14	Kirkland St. and Oxford St.
	15	Kirkland St. and Irving St.
	16	Boylston St. at Power House.
	17	Brattle St. and Ash St.
	18	Shepard St. and Walker St.
	19	Shepard St. and Garden St.
	20	Garden St. and Concord Ave.
	21	DeWolf St. and Cowperthwait St.

TABLE V

Key to Location of Samples, Analyses of which appear in Diagram 2.

Determination	Sample Number	Location
Total Solids on Streets with Car Lines.	1	Mt. Auburn St. and Maynard Pl.
	2	Cambridge St. and Maple Ave.
	3	Hampshire St. and Elm St.
	4	Hampshire St. and Plymouth St.
	5	Hampshire St. and Amory St.
	6	Hampshire St. and Tremont St.
	7	Mechanics Square.
Total Solids on Streets without Car Lines.	8	Elmwood Ave. and Fresh Pond Parkway.
	9	Brattle St. and Fresh Pond Parkway.
	10	Brattle St. and Craigie St.
	11	Brattle St. and Ash St.
	12	Brattle St. and Fayerweather St.
Silica on Streets with Car Lines.	1	Mechanics Square.
	2	Hampshire St. and Plymouth St.
	3	Hampshire St. and Elm St.
	4	Hampshire St. and Amory St.
	5	Cambridge St. and Maple Ave.
	6	Mt. Auburn St. and Maynard Pl.
	7	Hampshire St. and Tremont St.
Silica on Streets without Car Lines	8	Brattle St. and Fresh Pond Parkway.
	9	Fresh Pond Parkway and Elmwood Ave.
	10	Brattle St. and Ash St.
	11	Brattle St. and Craigie St.
	12	Brattle St. and Fayerweather St.

TABLE VI

Key to Location of Samples, Analyses of which appear in Diagram 3.

Determination	Sample Number	Location
Total Solids on Streets with Car Lines.	1	Garden St. and Appian Way.
	2	Mt. Auburn St. and Elmwood Ave.
	3	Mt. Auburn St. and Aberdeen Ave.
	4	Broadway and Ellsworth Ave.
	5	Cambridge St. and Quincy St.
	6	Massachusetts Ave. and Martin St.
	7	Regent St. near Railroad.
	8	Western Ave. and Hews St.
	9	Hampshire St. and Elm St.
	10	Kendall Square.
	11	Massachusetts Ave. North of M. I. T. Bldgs.
	12	Mechanics Square.
Total Solids on Streets without Car Lines.	13	Beaver St. and Cowperthwait St.
	14	Quincy St. opp. Colonial Club.
	15	Holmes Pl. and Peabody St.
	16	Brattle St. and Hawthorne St.
	17	Magazine St. and Erie St.
	18	Linnean St. and Raymond St.
	19	Arlington St. and Washington Ave.
Silica on Streets with Car Lines.	1	Mt. Auburn St. and Elmwood Ave.
	2	Broadway and Ellsworth Ave.
	3	Regent St. near Railroad.
	4	Mt. Auburn St. and Aberdeen Ave.
	5	Cambridge St. and Quincy St.
	6	Mechanics Square.
	7	Western Ave. and Hews St.
	8	Mass. Ave. North of M. I. T. Bldgs.
	9	Hampshire St. and Elm St.
	10	Massachusetts Ave. and Martin St.
	11	Garden St. and Appian Way.
	12	Kendall Square.
Silica on Streets without Car Lines.	13	Magazine St. and Erie St.
	14	Beaver St. and Cowperthwait St.
	15	Brattle St. and Hawthorne St.
	16	Quincy St. opposite Colonial Club.
	17	Linnaean St. and Raymond St.
	18	Mt. Auburn St. and Elmwood Ave.
	19	Arlington St. and Washington Ave.
Iron on Streets with Car Lines.	1	Hampshire St. and Elm St.
	2	Kendall Square.
	3	Regent St. near Railroad.
	4	Garden St. and Appian Way.
	5	Western Ave. and Hews St.
	6	Cambridge St. and Quincy St.
	7	Massachusetts Ave. and Martin St.
	8	Massachusetts Ave. North of M. I. T. Bldgs.
	9	Mt. Auburn St. and Aberdeen Ave.
	10	Mechanics Square.
	11	Mt. Auburn St. and Elmwood Ave.
	12	Broadway and Ellsworth Ave.
Iron on Streets without Car Lines.	13	Arlington St. and Washington St.
	14	Magazine St. and Erie St.
	15	Linnean St. and Raymond St.
	16	Brattle St. and Hawthorne St.
	17	Beaver St. and Cowperthwait St.
	18	Holmes Pl. and Peabody St.
	19	Quincy St. opposite Colonial Club.

CCCLXX

THE PROBLEM OF CITY DUST

BY REGINALD PELHAM BOLTON, MEMBER

The presence of dust in the atmosphere is practically universal. Its effects are far-reaching and extend even into the upper regions of the world's atmospheric envelope. Light and humidity are affected by its presence, as well as the comfort, convenience and cost of human existence.

The experiments of Aitken, in 1880, established the fact that the condensation of moisture into the form of fog and mist was primarily due to the existence of floating dust.

Samples of air from which dust had been carefully extracted were found to remain in a super-saturated condition without condensation, but this condition changed instantly to fog or mist upon the introduction of dust into the air. The particles of dust appear to act as nuclei around which moisture gathers, but in the absence of such opportunities of deposition of its moisture, air will not part with its contents of humidity. The conclusion arrived at was, "the larger the amount of dust in the air, the greater the tendency to form fog and rain." The haze and mist which envelopes large cities was shown by observations to be accompaniments of the dust arising from them.

These observations indicate the importance of the subject of dust, in its bearing upon the humid and misty conditions, and the prevalence of fogs which afflict great centers of population.

A contributory element of temperature in the combination of humidity and dust is evidently due to the heat generated by the concentrated combustion of fuel within the area of a great city, which increases the capacity of the superincumbent atmosphere for the absorption of moisture. Aitken's observations showed that the transparency of the atmosphere increases with its dryness. When the wet bulb depression is 8 degrees, the transparency is 3.7 times greater than when the depression is only 2 degrees.

In considering such conditions as prevail in this respect, in the two greatest centers of population, it is to be borne in mind that the congestion of population in New York is much greater than it is in London. In the latter, the total metropolitan area is 121 square miles, within which the most densely populated portion of about 21 square miles was estimated by Fitzmaurice to contain 1,300,000 people. In New York, the area of the Island of Manhattan is about a similar extent or 21.93 square miles, but the population of that borough exceeds 2,200,000 persons, or 100,000 to the square mile, or one person to every 279 square feet. The effect of such concentration upon the emission of heat, and of the generation of dust must, therefore, be very marked.

The peculiarly evil character of the fogs from which London has so long suffered has generally been attributed to the presence of the smoke emitted by its multitudinous domestic fires of bituminous coal without special regard to the contributory elements of heat and especially of dust. Public attention there as well as in this country has concentrated on the subject of smoke abatement as a remedy and the literature of that question during the past fifty years would fill a considerable library. An enormous amount of study, effort and expense has been devoted to the elimination of visible smoke, but the trend of modern investigation in this regard should now be towards the elimination of the several elements that go to make up what is commonly regarded as "smoke."

Blackened smoke, which is commonly regarded as the sole cause of offense, is merely the same element which is being emitted by apparently smokeless chimneys, but it is colored by unconsumed tar, and darkened by unconsumed cinder and ash. Bituminous or "soft" coal, has about 100 pounds of tar to the ton weight, and this material is the most difficult to bring to a state of complete combustion under the variable conditions of furnace design and temperature.

Cary gives the following analyses of the composition of solid materials emitted with furnace gases or smoke, by which the proportion of the discoloring materials of tar and sulphur are shown:

THREE TESTS OF SOLID MATTER IN COAL CASES—A. A. CARY

Carbon	86.8	66.25	77.26
Hydro Carbon and Tar	4.35	4.82	3.69
Sulphur	0	3.75	1.7
Ash, etc.	8.85	25.2	17.35

But all smokeless chimneys emit their gases of combustion, and though the volume is less with anthracite coals, the ash content is larger, and a stronger draft is necessary for its combustion. New York is the largest centre of consumption of anthracite and so the subject of the cinder and ash emitted from its chimneys becomes of direct interest; especially if such materials bring in their train, the same fogs, humidity and mists which afflict many other cities.

The investigations of the Smoke Abatement Society in London established some fairly definite statistics as to the quantities of solid material which are emitted from the bituminous coal burned in that city, and which find a place of rest within its area. On an area of 117 square miles, there was deposited 76,050 tons of soot, or 650 tons per square mile per annum.

As an illustration of the effect of density of population and corresponding coal consumption, observations taken outside of the metropolitan area, in the neighboring county of Surrey, showed that this deposit fell to the rate of only 195 tons per square mile per annum, or less than one-third within the city. Clearly, therefore, the rate of deposit is connected with the amount of domestic fuel consumed.

Analyses which were made of the materials thus collected indicated its direct connection with fuel combustion. It included the following:

Carbon and tar..................	59,000 tons
Sulphates	8,000 tons
Ammonia	6,000 tons
Chlorides	3,000 tons

Some efforts were made in Chicago, in 1910, to determine the amount of material emitted by locomotives operated within the city. The report of the Chief Smoke Inspector states that from 8 to 18 per cent. of all fuel fired in a locomotive was found to be discharged in the form of cinders, and as about 1,400 locomotives were at work within the city limits, it was estimated that at an average rate of ten per cent. of the fuel consumed, 560 tons of cinders per day were spread over a limited part of the city. The Department of the Smoke Inspector made an attempt to secure some data similar to those obtained in London, by placing pans at various places in the City, but the observations were abandoned owing to the pressure of other work.

Tar blackened smoke is undoubtedly a source of direct loss to a community.

The cost to the public, in one way or another, which results from the deposit of discoloring materials emitted by chimneys, was estimated by W. Meng,to amount to fifty millions of dollars a year, though this can be, at the best, only a guess at its extent.

Upon such items as the cleaning of the exterior of city buildings, considerable expenditures are frequently required. Such cleaning costs from $500 to $5,000, according to their size and the material composing their exterior surfaces. The frequent repetition of such expense, evidently depends directly upon the fuel conditions of their location. Sulphur is another constituent of chimney gases very destructive of the color of certain building materials.

In all cities, and even in their vicinity, a large proportion of all domestic expense, especially of domestic labor, is attributable to the effects not only of smoke but of dust. The reduction of sunlight, especially in the early part of the day, is very marked in certain cities afflicted with smoke products.

But apart from the discoloring elements of tar and sulphur, it is evident that any reduction in the enormous volume of dust of large cities would confer widespread benefits upon such crowded communities. The heating engineer can deal with one of the largest, if not the largest, element in this infliction.

The elements composing city dust doubtless vary greatly in various localities, but there can be little doubt that a substantial proportion is usually contributed by fine ash and cinder emitted from chimneys. Such materials are not the most deleterious from a health standpoint, since they must be free from organic contamination, but it seems that they constitute a substantial part of the volumes of dust which afflict city existence, and add to the expense of street cleaning and of every domestic establishment.

No definite measurements of deposits of similar character to those made in London, have so far been carried out in New York. It is, however, easily to be seen that the amount of such material pendent in the superincumbent atmosphere may be very considerable and the effects proportionately far-reaching. Since ash and cinder is seen to be the largest element in city dust, the statistics of our consumption of fuels bear on these proportions.

In New York City and harbor, nearly twenty millions of tons of coal are annually consumed, of which about fifteen millions of tons were consumed directly within the city. On the basis of the emission, as cinder and ash of only one-half of one per cent. of its total weight, this fuel would deposit over the closely occupied area

of the city, or 130 square miles, five hundred and seventy-five tons per square mile per annum.

"The dust of cities," says Dr. Soper, "is composed of par-"ticles of every conceivable substance which is capable of being "ground into a finely divided condition by wear and tear, among the "innumerable activities of the people."

"Little care is taken to prevent dust from polluting the atmos-"phere. Carpets and rugs are shaken on house steps, sweepings "are thrown from windows, and dust composed largely of horse-"manure lies in windrows on sidewalks and roadways."

"In nearly every large city there are factories in which large quan-"tities of harmful dusts are produced. Building operations, over-"loaded coal and refuse carts, push carts and horses also contribute "largely to the production of city dust."

"Calculations on the consumption of horseshoes, the tires of vehi-"cles, and the brakes, rails and machinery of street and elevated "cars, show that the amount of iron and steel ground up every day "in the heavy and congested traffic of New York is enormous."

To this it may be added that the dust near or on the ground, which specially contributes to street accumulations and thus contaminates the air inhaled by the population, includes a large proportion of cinder and ash, ground to impalpable powder by foot and road traffic.

The Lederle Laboratories had occasion a few years ago to examine street sweepings taken from various parts of the city. These were analyzed as follows, and showed a proportion of ash content which varied according to the season of the year and was largest in the heating period:

Month.	Per cent. of Ash.
March 17	85.06
April 15	80.03
June 6	57.02
June	58.44
August 21	42.26
August 27	20.39
September 13	29.77
September 25	74.14

"The most striking feature of these sweepings," says Dr. Joseph A. Deghuee, of the Lederle Laboratories, "is their large proportion of ash or mineral matter."

"The average of 8 samples of street dust taken from different "points in Manhattan, late in September, shows nearly 75 per cent. "of ash."

"It may be that some part of this ash could be due to abrasion "of pavement surfaces, but consideration of the circumstances leads "to the conclusion that nearly the whole must be fine ash and ground "cinder emitted from chimneys."

The area of a New York street surface relative to the built-up or occupied portion is about 44 per cent. Buildings and their courts and yards thus receive about 56 per cent. of all descending ash and cinder. It is from such sources that the advocates of "fresh air" or "open window" ventilation would provide the occupants of buildings with their supply of air for respiration.

The concentration of dust at and near the level of street, which such ill-considered methods would introduce into our school-rooms, is indicated by observations recently made by Prof. Whipple, at the Woolworth Building. The number of dust particles by count at the sidewalk level was twice that at a height of a story above the street, and ten times the number found in the air at the upper levels of this tall tower building, but at the street sidewalk, the count showed over 200,000 particles to the cubic foot. Samples of air taken at the same date, on a steamer on the Sound, showed only 18,000 particles. These are merely counts of the number of particles, and afford no information as to their nature.

It is a matter of scientific knowledge that the fine ash and cinder which is expelled by volcanic explosions to enormous heights, is, by the action of the winds, drifted to immense distances over the surface of the earth. "Heated air in the tropics carries organism-charged dust in great volumes into the higher air, finally to settle everywhere."

Recent examination has disclosed the presence even in such volcanic dust of organic materials, and it is stated that bacteria have been found therein, possibly due to the admixture of other dust carried to great heights, but even cosmic dust when subjected to analysis was found to contain bacterial elements.

A similar, if less far-reaching effect is produced by the vast volume of heated gases rising from great cities. The cinder and ash emitted from their chimneys are thus liable to be spread by the motion of the wind over a very considerable area in their vicinity.

But unfortunately such mere inert material is not the only content of city dusts.

Bacteria are vastly more numerous in the air of cities than in the atmosphere of the open country.

Among the worst ingredients of city dust are the parasitic bacteria. The most harmful are produced by the human body.

Dr. George A. Soper, in his work on Subway Ventilation, gives the following data on Bacteria in City Dust:

A Broadway Theatre	270,000 per gram.
New fashionable hotel	360,000 per gram.
Fifth Avenue Church	320,000 per gram.
Large office building	850,000 per gram.
Subway	500,000 to 2,000,000

"The Kingdom of Dust is the Kingdom of Death and Sanitation "must declare war against the Kingdom of Dust."

Such are modern expressions of the views of medical authorities, published in "The Medical Council," drawing attention to a condition nearly universally prevalent, and especially marked and menacing in the case of crowded communities. Dust is placed in company with flies, as an equal menace to health, in the recently formulated rules of the Department of Health in New York City.

The largest single cause of mortality in New York City is tuberculous and bronchial disease, a condition directly connectible with the prevalence of dust. The common liability to catarrh among city residents is a result of the presence of excessive dust.

The contribution to these dangers to health by the presence of dust, which is made by the combustion of fuel, is not in the direction of the organic constituents but inasmuch as all dust is deleterious to health it is actually one of the large elements in the general menace to public health and life. City dust, and its elimination, therefore, forms a subject of practical and economic value. It should be of peculiar interest to those engineers who are engaged in designing and operating steam plants for power, lighting and heating, which involve in the aggregate the combustion of enormous quantities of fuel and the emission of a proportionately large addition to domestic and municipal costs.

Such considerations as these indicate that the work of the ventilating engineer should always include the elimination of dust.

The true principles of hygienic ventilation cannot be found in any method of uncleaned air circulation, and the open-window and so-called fresh air methods of ventilation mean the admission of dust, discoloring and damaging property, and deleterious and dangerous to health.

CCCLXXI

CINDER REMOVAL FROM FLUE GASES OF POWER PLANTS

BY C. B. GRADY*

When coal is burned in a furnace, small particles of unconsumed carbon, dust and ash, are carried away from the fuel bed and a portion of these small particles, commonly called cinders, are drawn up the chimney and thrown out into the atmosphere.

These cinders cause a considerable percentage of the dust in the atmosphere of our cities. They penetrate into buildings through windows and ventilating ducts and their presence is a source of annoyance and expense on account of their tendency to smudge and discolor light colored materials.

The mitigation of any portion of this affliction to city existence, is of direct interest to engineers working on the problem of ventilation, and it is the object of this paper to describe and illustrate several large installations of cinder and dust catchers for washing the gases coming from boilers. The author believes that the same general principles which have been so successfully applied to the large plants, could be adopted to most all installations, whether large or small.

The fuel consumption at the Waterside Stations of The New York Edison Company is at the rate of over 600,000 tons of fuel per annum. The local effects of such a concentrated operation are intensive, and the dust and cinder emitted from the stacks were a cause of annoyance to those in the neighborhood. Public complaints of these effects were made while the Company was devoting special attention to the elimination of the cinder difficulty, and during the years 1906 to 1912 trials of various suggested methods and apparatus were made, all of which failed to produce effective results. But by a process of elimination these disposed of all the suggested methods and led the way to the development and successful adapta-

*Non-Member

tion to the problem of the Metropolitan cinder and dust catching apparatus which is installed on 150-650 horse-power B. and W. boilers.

Figure No. 1 shows a cross section of the flue, the cinder and dust catchers for two rows of boilers in Waterside Station No. 2, of the New York Edison Company, and the cross section of one of the boilers located at the left hand side of the flue. The movable damper "D" is about 5 feet high and 50 feet long, running the entire length of the flue. The water tank at the bottom of the flue is 8′ 6″ wide by 50 feet long, and about 18″ of water is maintained in the tank. Water is taken from the tank and pumped into the water pipe "K" by means of a small lowhead centrifugal pump. The water flows out of the pipe "K" through a number of 1″ holes, spaced about 4″ center to center, into the gutter "G," then flows over the edge of the gutter "G" down the inside face of the damper "D" into the tank. About 75 gallons of water per minute per boiler are thus circulated, and about eight gallons per minute per boiler are added to make up for the evaporation and for the water spray which is carried away by the gases. Salt water is used at the Waterside Stations, the water being taken from the fire service. The gases originally left each boiler at the rear through a horizontal intake "A" and traveled along a horizontal rectangular flue "B" to the stacks. The new arrangement with the cinder catchers is as follows: The gases leave the boiler through the same horizontal intake "A" and are deflected downward by a baffle plate "C" and then pass down through a wedged shaped duct, one side of which is formed by the movable damper "D" and the other side by one of the sides of the flue. The above mentioned duct is of uniformly decreasing cross section and is open at the lower end, so that the direction of flow of the gases as they leave the duct is substantially vertical. In passing through the cinder and dust catcher the gases are fanned out, so that they leave the bottom of the catcher through a narrow, long slot. The water level in the lower portion of the flue is kept constant and at a comparatively short distance from the bottom of the damper "D." The water flowing into the gutter "G" and running over the edge of the gutter forms a sheet of water on the damper "D" which makes a little waterfall from the bottom of the damper "D" to the water in the bottom of the flue.

The cinders and dust are caught either by coming in contact with the sheet of water on the damper "D" and thus being carried into the water below or are projected into the water in the bottom of the flue by the comparatively high velocity that they have attained.

This velocity is attained partly by the increase in the velocity of the gas current in which the cinders and dust are suspended and partly by gravity. This is an important feature, for the inertia of

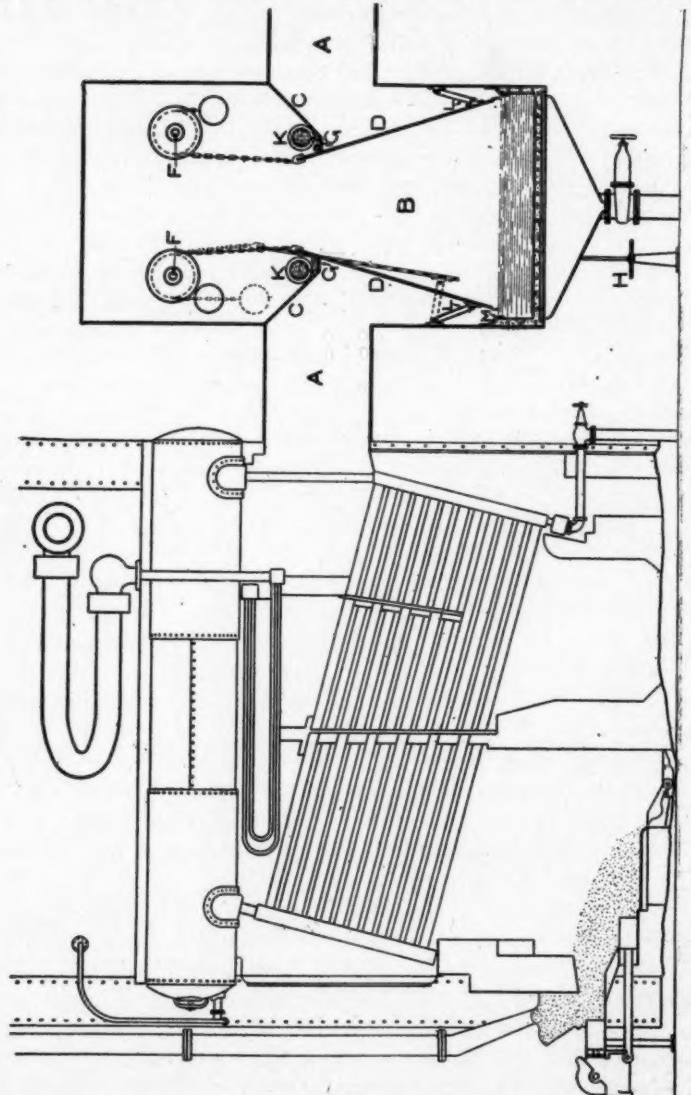


Fig. 1—Section Through Boiler, Flue and Cinder Catcher, Waterside No. 2 Station.

the solid particles varies in proportion to the square of the velocity and the resistance of the projected particles to any force tending to change their direction of travel varies as the inertia. It is estimated that the cinders and dust are projected toward the surface of the water with a velocity of about 50 feet per second.

Referring to Figure 1, it will be noted that the two positions of the damper "D" are shown. The position showing the damper "D" in full lines is used when the boilers are operating at light loads and the position showing the damper in dotted lines is used when the boilers are operating at a maximum load. The dampers are raised and lowered by turning the shaft "F" which is operated from the boiler room floor by means of a hand wheel "H." The lower portion of the movable damper "D" is connected to the side of the flue by means of a link "L" so that the bottom of the damper "D" will move upwardly and outwardly when the top of the damper is raised, and maintain the distance from the side of the flue to the bottom of the damper and the distance from the surface of the water to the bottom of the damper approximately equal. Fig. 2 is a view of the inside of the cinder catcher.

A serious problem was encountered in the operation of the catcher, due to the corrosive action of the acid water on the baffles. The gases leaving the boilers carry with them an amount of sulphur dioxide gas, varying with the percentage of sulphur in the coal. This gas has a decided affinity for water and a dilute solution of sulphurous and sulphuric acids is formed. This acid production is accelerated by the high temperature of the flue gases. When salt water is used some hydrochloric acid is formed as well. In cases where it is necessary to recirculate the water a number of times the acid solution becomes concentrated to some extent and it was exceedingly difficult to find a commercial metal that would withstand the corrosion. A sheet iron baffle, for instance, was eaten away in two weeks. Then enameled sheet iron was used but the enamel cracked off as a result of the expansion of the metal when subjected to the heat of the flue.

Asbestos board was then suggested as a material that would possibly meet the condition. Experiments showed, however, that while it resisted the acid and heat well, it became water-soaked and soon softened and deteriorated. Various special alloys with high silicon content were tried but they cracked after a few hours service and were abandoned. Copper sheets and cast iron plates resist the action the best of any metals that have been tried but even these are slowly attacked.

Glass is affected neither by acids nor by water and some baffles were made of wire glass but it was found that the heat strains were too severe and the baffles went to pieces very quickly. A hard glass is too brittle to withstand the heat, and breaks, while a soft glass softens under its action and warps.

The material of which the baffles are made must possess sufficient mechanical strength to permit them to be made of considerable length and to withstand the stresses and shocks incident to their

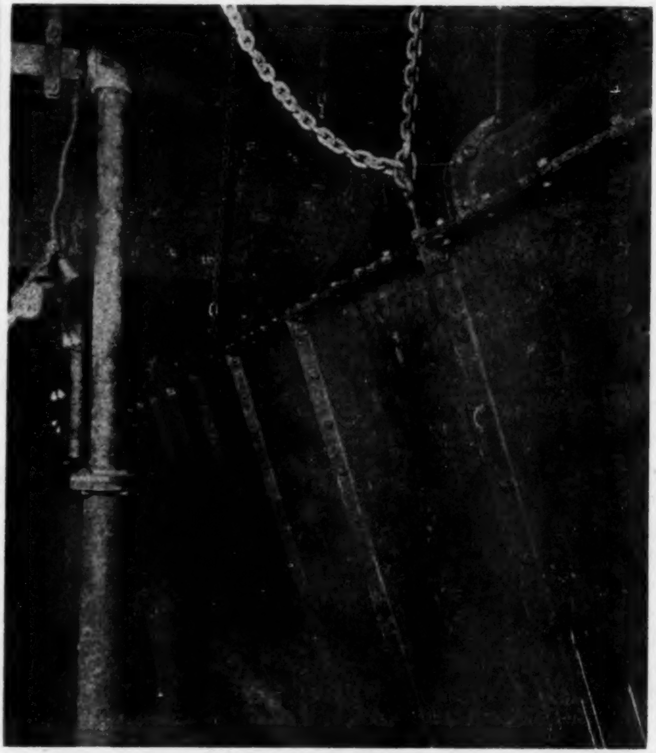


Fig. 2—View of Inside of Cinder Catcher, Waterside No. 2 Station.

operation and yet be light enough so that the baffles can be easily moved; it must have temperature characteristics which will prevent it from either warping or cracking under the influence of the heat of the gases at the flue temperatures; and it must further be non-corrodible. The last requirement is a difficult one to meet, for while

the acid is a very dilute one, yet the washing action of the water flowing down the baffles constantly removes the salt resulting from the chemical action and presents a fresh surface for attack.

Trials have been made with baffles made of cypress lumber and these have given good service. They are light yet strong, are

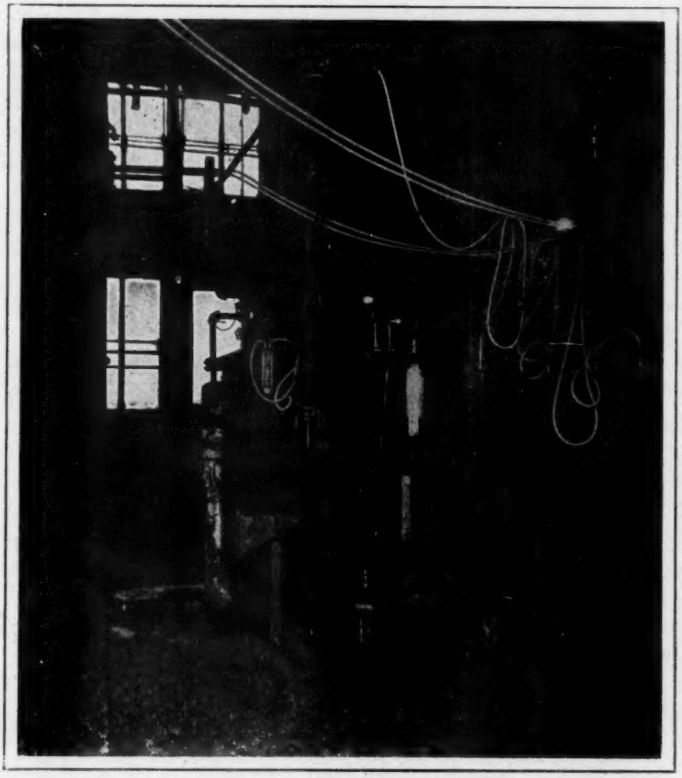


Fig. 3—General View of Testing Apparatus.

unaffected by acid or water, and will stand the heat as long as water is kept flowing over the surface. In addition they are cheap and can be readily replaced or repaired when necessary. Their only limitation is that water must be kept on them at all times that the boilers are in service.

The most satisfactory service, however, up to the present time has been obtained with baffles constructed of concrete. These con-

sist of cinder concrete slabs about one inch thick reinforced with expanded metal.

To determine the effectiveness of the catcher in cleaning the gases, and for the purpose of comparison, simultaneous runs were made on a stack provided with a cinder and dust catcher and one that was not so equipped.

A great deal of time and care was expended in perfecting apparatus for testing purposes that would give accurate results. A general view of the apparatus used at Waterside is shown in Fig. 3.

The general procedure of the tests was as follows: A portion of the gas was drawn from the stack through a sampling pipe by means of a steam jet. The solid matter in the sample was weighed and proportioned to the total volume of gas passing up the stack in the ratio of the areas of the stack and the sampling pipe. The pounds of coal burned while the sample was being taken was found by counting the revolutions of the stokers during that time. The total weight of solid material going up the stack was then divided by the coal burned to reduce it to a unit basis.

The relatively small portion of gas drawn from the stack through the sampling tube, made necessary the securing of a sample that would be truly representative of the average composition of the stack gases, since the whole weight of cinder and dust emitted is based on this sample. It is an established fact that if gases containing solid matter in suspension are drawn through pipes from mains at a higher velocity than that at which the gas is moving in the main, a lower percentage of solid matter will be found in the gas than actually exists, and vice versa, a large quantity will be found if the suction produces a less velocity than the velocity of the gases in the main. In view of this it was necessary to regulate the flow of gas in the sampling pipe to a velocity equal to the average velocity in the stack and also to locate the end of the sampling pipe at a point in the stack where the average velocity existed.

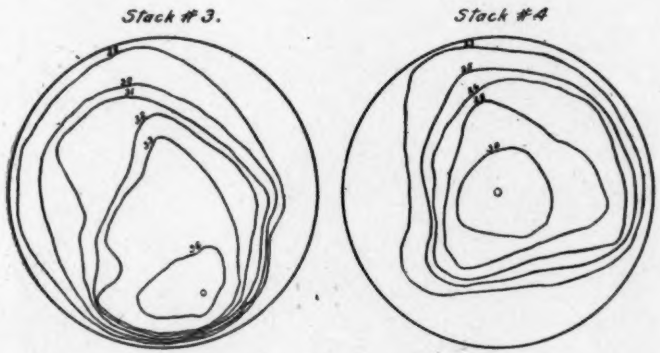
The average velocity was determined by exploring the stack with a triple Pitot tube. This determination disclosed some very interesting phenomena as shown by traverses of stacks Nos. 3 and 4. figures 4 and 5.

In stack No. 4 the velocity varied from 23 to 30 feet per second, and the point of maximum velocity was close to the center, as might be expected, but in stack No. 3 the velocity varied from 28 to 36 feet per second, and the point of maximum velocity was near the edge of the stack. Another remarkable feature disclosed was that under normal conditions these traverses remained substantially con-

stant for a period extending over two weeks, during which the outside atmospheric conditions, such as humidity, temperature, pressure and velocity of wind varied through rather wide ranges.

At periods of low loads still more remarkable variations in the velocities developed, and for a certain area near the edge of stack No. 4 the Pitot tube gave a minus reading, indicating a down draft. That this down draft existed was proven by introducing small pieces of paper which were carried down for twenty feet or more, and then were caught by a swirl or eddy current and rapidly carried up the opposite side of the stack. Pieces of paper inserted at another point hung suspended, showing the absence of any movement of the gas whatever.

The above interesting phenomena show how difficult it is to theoretically figure the flow of large volumes of gases at low velocities, and these and other experiments along similar lines may possibly lead to a change in our formulas for the design of flues, uptakes and stacks.



Figs. 4 and 5—Stack Traverses.

The results of the tests made at Waterside indicate that the smoke in a stack operating under ordinary conditions contains cinder and ash amounting to from 1% to 2% of the weight of coal fired. Of this material the cinder and dust catcher is capable of removing 95%. This highly gratifying result led to the complete equipment of both Waterside plants with this type of apparatus. The effects on the neighborhood are evident, the sidewalks in the immediate vicinity of the station are no longer covered by a cinder deposit and complaints from the neighbors have ceased entirely.

Other Central Stations in the city with the same trouble to overcome have recently installed the apparatus.

At the Gold Street Station of the Edison Electric Illuminating Company of Brooklyn, New York, there are in service sixteen cinder and dust catchers which wash the gas coming from 16-650 horse-power B. & W. boilers. A photograph of one of them, with a second appearing in the background, is shown in figure No. 6, and a section of a boiler and cinder catcher is shown in figure No. 7. The cinder catchers for the Gold Street Station are similar in principle to those heretofore described for the Waterside Stations, but they are constructed entirely of cast iron plates ⅞" in thickness. The cast iron, after about a year's service, appears to be withstanding the acid action very well.

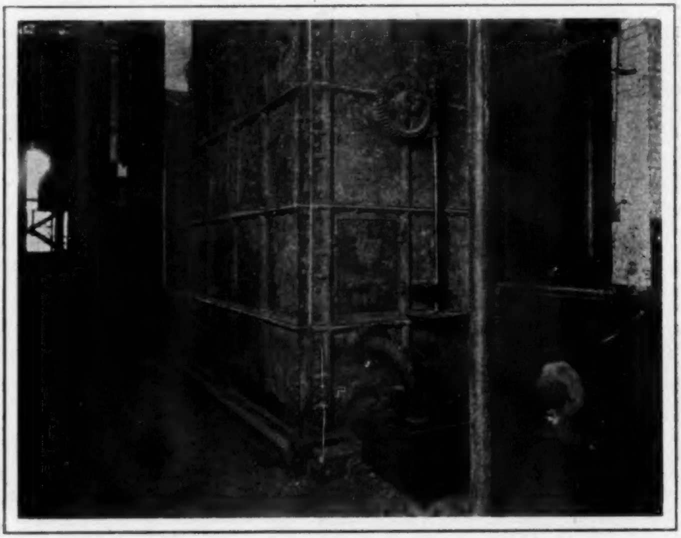


Fig. 6—General View of Cinder Catchers, Gold Street Station.

Referring to figure No. 7, the path of the gases is as follows: The gases leave the boiler and enter the cinder catchers through the flue "A" and are deflected downwardly toward the surface of the water in the base of the cinder catcher by the stationary baffle plate "C" and the movable damper "D." The movable damper "D" is shown in its lowest position, but it may be raised to the position indicated by dotted lines when the load on the boiler is increased. A sheet of water may be maintained on the baffle plate

and damper by means of cast iron pipe "K" which is provided with 1" holes placed about 4" centre to centre. The gases leave the cinder catcher through the up-take "B" which leads to the stacks.

Thirty-two boilers at the 201st Street Station of the United Electric Light & Power Company are equipped with Metropolitan cinder and dust catchers. The boilers are similar to those at the Waterside Stations and are located on one floor, laid out in four rows of eight boilers each. Two rows are placed on each side of the boiler room, back to back, thus making three firing aisles. The gases from the boilers pass through the vertical up-takes which, in turn, discharge into the stacks. A cinder catcher is installed in the base of each up-take and figure No. 8 is reproduced from a photograph of the lower portion of one up-take.

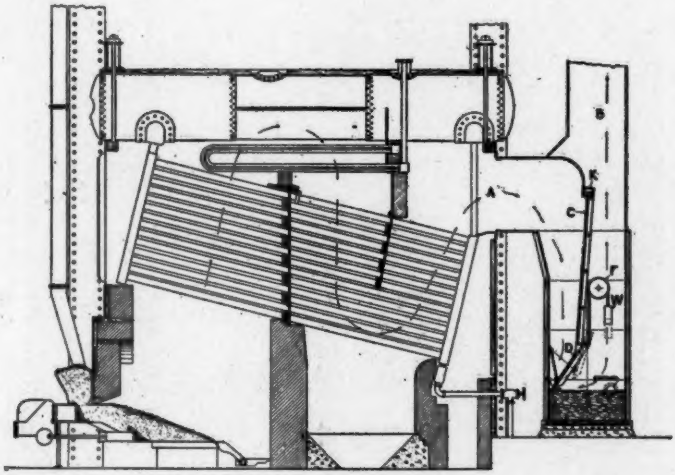


Fig. 7—Section Through Boiler, Flue and Cinder Catcher, Gold Street Station.

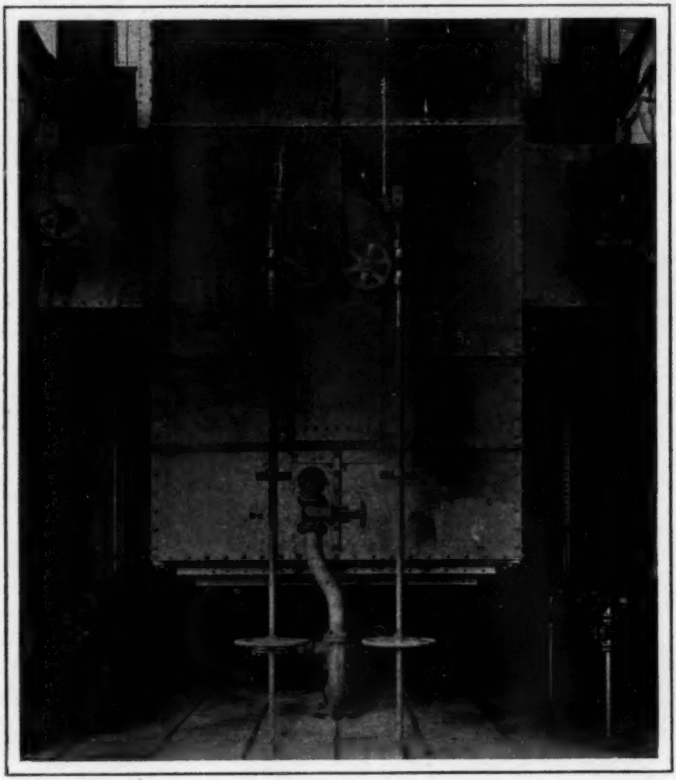


Fig. 8—General View of Cinder Catcher, 201st Street Station.

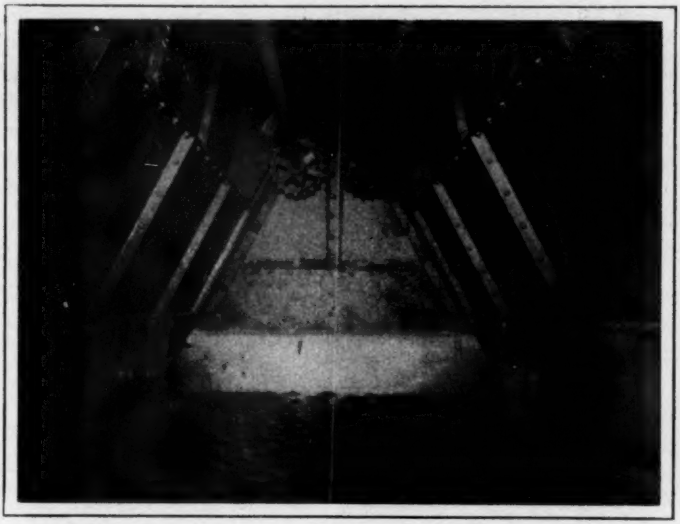


Fig. 9—View of Inside of Cinder Catcher, 201st Street Station.

Figure No. 9 is an interior view showing the arrangement of the movable dampers and operating mechanism. The dampers are raised and lowered by turning the hand wheels on vertical shafts shown in figure No. 8.

DISCUSSION

At the annual meeting three papers were presented relating to the problem of city dust. One was by Profs. C. C. and M. C. Whipple, "Studies in Air Cleanliness"; one by Mr. R. P. Bolton, "The Problem of City Dust," and one by Mr. C. B. Grady, "Cinder Removal from the Flue Gases of Power Plants." They were all discussed at the same time.

Mr. D. M. Quay: The paper by Mr. Bolton is certainly a very valuable paper. I was hoping that the writer had gone further and given us some methods of preventing first, the soot and other products from the chimneys being distributed to the air and second, a practical method of removing the dust and other impurities. I am convinced that a vacuum cleaning system would be a great improvement over the present method of disturbing the dust particles and having them distributed through the air. I think there may be some

very practicable and instructive points brought out as a result of Mr. Bolton's excellent work.

Mr. Baldwin: I desire to ask Mr. Grady what percentage of the total chimney dust is taken out by the process referred to in his paper?

Mr. Grady: 95%.

Mr. Bolton: Mr. Grady's paper brings before us in a very definite form the remedy for the difficulty which I set forth in my paper. The problem which Mr. Grady has so successfully met is one of the greatest in the country.

One of the most interesting features demonstrated is that the amount of cinder from the boilers varied according to the rate of combustion. The cinder removed from the flue gases weighed from 1 to 2% of the coal fired, showing a greater percentage of ash and cinder in proportion to the draft that is required.

The material that is secured contains about 60% of carbon, and, therefore, has distinct fuel value.

Another point to which the paper draws our attention is the remarkable variations in the flow of gas in large stacks.

In solving the difficulties experienced in maintaining materials in the presence of the furnace gases, we ought to congratulate the engineer on his persistence. The simple conclusion was reached that a piece of wood might do what expensive metals could not accomplish.

Prof. Whipple's studies are a valuable contribution to our source of information on this subject. Prof. Whipple took his samples from a level above the street, and, therefore, secured in the receptacles either the descending or rising dust. Imagine, therefore, the condition upon the level of the street. Dr. Soper has shown in his investigations that the greatest contamination is on the sidewalks. In the subway, this is the air we are receiving. The so-called system of ventilation applied to our subways is a farce and a scandal. I think it is time that the attention of the public should be drawn to the fact that this condition is a public injury and one which in all new subway construction, should be cared for. Our subways are air sewers.

Mr. Baldwin: An apparatus I use to take dust out of the air is by the dry method. If anyone is interested and will be in town on Saturday, I will give him an exhibition of the apparatus in my office, with dust gathered from the roof, in the neighborhood of Riverside Power House.

Mr. Carter: I would like to ask if temperatures of flue gases after passing through water are greatly affected by the water.

Mr. Grady: The reducing of the temperature through the cinder catcher amounts to from 50 to 60 degrees. For instance, gas might enter at a temperature of 500 degrees and flow out at 450 degrees.

Mr. Davis: In Prof. Whipple's paper he speaks of oil and macadam roads. I have found a number of instances in New York where streets are oiled that we are coming into something entirely new in dust. The oil soaked dust permeates the atmosphere. A large cleaner informed me that since they began to use oil on streets the life of draperies has been cut in half and practically two cleanings are the end of them. The oil keeps the dust down for a short time, but these oil soaked particles of dust you will find all over the city. On Riverside Drive you will find places where it has gone up to windows and turned white trim into a dirty yellow. I don't think we want to go on record as stating that oil is a dust preventative.

Mr. Hart: It might also be interesting to tell of some results found in Chicago. They took some sweepings from Michigan Avenue, analyzed them and found a very large percentage of rubber.

CCCLXXII

DEVELOPMENT IN HEATING AND VENTILATING INDUSTRIAL BUILDINGS

BY E. L. HOGAN, CHAIRMAN.

When our great-great-grandfathers came to this country and began to practice their trades, each worked independently in his own little shop. Each had his crude tools and manufactured individually his particular article, whether shoes, chairs, or clothing, from the raw material to the finished product.

Naturally, their heating systems were not elaborate as such were not necessary. Usually a grate sufficed to heat them and the same grate was used to roast their meats.

With the development of American ingenuity the methods of manufacturing gradually changed. The next generation conceived the idea of specializing; finding that if they performed one certain portion of the work of manufacturing they became more expert at this part and were enabled to earn more money. The next generation specialized even more, and so on, until, to-day, our industrial world is specialized into a number of large organizations, each making some one particular portion that may be assembled into a finished product.

Through this development methods of heating naturally became more elaborate. The old-fashioned fireplace developed into a stove. The need of a centralized heating system developed the furnace placed in the cellar with pipes leading to several rooms, which were heated by warm air. The use of steam for power purposes called for further ingenuity in utilizing this same steam for heating the building.

The industrial building of to-day is heated by one of three methods: direct steam; direct hot water; or by forced circulation of air which is heated by heaters, supplied by steam or hot water, and centrally located.

The greater number are heated by direct steam because it can be installed by any steam fitter and frequently the superintendent with

his millwright and other help can do the work. They know more about this system than any other and are not keen for methods they do not fully understand.

In certain lines of industry where moisture from some drying process or otherwise, must be absorbed by the air within, the blower or hot blast system is most efficient because by means of it the moisture is removed by the circulation of the air that heats the building. Also in buildings where smoke or gases must be removed, the forced circulation is desirable.

In order to study the movement of air currents let us assume a building as illustrated in Fig. 1. This could be used as a foundry, machine shop, car shop, structural steel plant or forge. The side walls are of concrete blocks and glass set in steel sash; the frame work is of steel, the floor of concrete; the roof of reinforced concrete with composition roofing cover.

The natural air currents in the structure would be as indicated by small arrows in the illustration. This movement is caused by the difference in temperature of the air within; that coming in contact with the cold roof, monitors and outside wall becomes cooler and heavier, and consequently falls. The warmer air in contact with the floor rises and circulation is set up tending to equalize the inside and outside temperature.

From a heating standpoint we are interested in a strata covering the lower eight feet of the interior, this must be kept comfortable and fairly uniform. It can be accomplished by creating currents in the upper portion of the building so that no cold air will reach the floor, taking care of the outside walls by radiant heat from radiators placed above and out from the walls, or below and on or near the walls, and taking care of the lower strata of air by heat from the hot strata above. This method is practically keeping the exposed surface of the structure warm so that there will be no cold currents in the working portion.

In structures of this kind where the peak of the roof is 40 feet or 50 feet above the floor and a working temperature of 65 degrees near the floor is maintained, the temperature near the peak will be approximately 90 degrees. While it is apparent that by keeping the exposed surface warm or keeping a layer of warm air next to it, that the inside of the structure will be warm, nevertheless, it would seem as though it would be better to keep the working strata warm and do so in such a way as to prevent the cold currents from entering this strata. This would reduce the temperature of the air in proximity to the cold exposed surface and decrease the heat loss from the building.

In order to produce results like this it would be necessary to continually mix hot air with the air in the working strata in order to maintain a working temperature. In this system the hotter portion of the building is the lower, and the radiant heat is up and toward the exposed surfaces. Air mixes or diffuses very rapidly and sufficient air must be circulated in the lower working strata to maintain a uniform temperature. Such a system is what is aimed at with hot blast and what is frequently obtained. It is surprising how well and with what a little heating surface and consequent small steam consumption you can heat a building in this way.

A distributing system which would approximate these results is indicated on this plate in Fig. 2.

Frequently it is attempted to heat a structure like this with a hot blast system by simply having outlets in the mains and no drop pipes. In this case it would be necessary to direct air toward the outside walls and downward at various angles in order to produce a distribution and the heating would revert to a condition where you would have to warm or produce a layer of warm air next to the exposed surfaces and in this it would be the same as a direct radiation system. The exposure in this case would be larger and the usual method of figuring would not suffice.

In figuring the heat losses from a building you should first determine how you are going to heat it and then it will be possible to determine what the difference in temperature may be between the inside and outside of the various parts, such as outside wall, glass, roof, monitors, etc. As it is now, we take the temperature desired and assume that this is the inside temperature all over and with all methods of heating. The heat losses from the building shown in Figures 1, 2 and 3 would probably be determined as follows:

Inside temperature, assume 65 degrees.

Outside temperature, corresponding 0 degrees.

Exposures would be as follows:

Exposed wall surface.

Exposed wall glass surface.

Exposed roof.

Exposed monitor wall.

Exposed monitor glass.

Exposed floor.

Using constants for radiation losses our table of losses would be as follows:

Kind of Surface	No. of sq. ft.	Diff. in Temp.	Constant	Con. x Temp. diff.	Heat loss in B.t.u.
Outside wall	9,040	65	.33	21.45	194,000
Outside glass ...	10,760	65	1.2	78.	838,000
Outside roof	38,400	65	.3	19.50	711,000
Monitor wall	3,600	65	.4	26.	83,000
Monitor glass ...	4,800	65	1.35	87.8	422,000
Monitor floor	36,000	35	.31	10.85	391,000
					2,649,600

Total heat loss equals..........................	2,649,600 B.t.u.
Heated day time. only add 10%..................	264,900 B.t.u.
For infiltration of air on windy days add 10%..	264,900 B.t.u.
	3,179,400 B.t.u.

Total cubical contents of structure = 1,180,000 cu. ft.

With a condensation of ¼ per sq. ft. of direct radiation, this structure would require 13,200 sq. ft. or 1 sq. ft. for every 89½ cu. ft. of contents.

With direct radiation placed along the outside walls or suspended from the roof trusses the average temperature between the inside and outside of the exposed surface would be greater, the heat loss approximately 3,663,000, and the radiation required would probably be approximately 15,250 or 1 sq. ft. to 77.4 cu. ft., providing for the same extra 10 per cent. to heat up quickly in the morning where the building is used during the day time only and also for loss due to infiltration of air on windy days.

Now if a hot blast system were used with drop pipes so as to distribute the air over the working strata, the average loss from the building would in all probability be approximately 2,680,000 allowing for the same extra amounts under similar conditions. However, to partially offset this, is the loss in temperature of the air due to radiation from the distributing ducts. This amounts to approximately 8 per cent., varying with the shape of building or the extent of the distributing ducts. This would be equivalent to increasing the radiation loss by 8 per cent. which would bring it up to 2,900,000 B.t.u. Thus it would seem that by applying the heat to the area where it is required, we would save approximately 21 per cent. of steam required. This would only be possible with a blower system and with distribution to the strata required as indicated in Fig. 2.

If the drop pipes or ducts were stopped within 3 feet of the main duct and the air directed out toward the outside walls and down, the problem would become similar to the direct radiation installation and the losses would be between 3,179,400 and 3,663,000 which would be more nearly correct, and in this event the saving due to the use of a properly designed hot blast system would be approximately between 9 per cent. and 21 per cent.

Owing to the fact that a hot blast system can be overloaded and is also capable of forcing the heat to all parts of a structure under adverse conditions, the average purchaser expects a great deal more from it than from direct radiation and our experience has been that

a hot blast system has to do about 20 per cent. better work than would be expected from a direct job. A great deal more is expected of something which is more conspicuous and which has a moving or operating part.

There is one thing which should be kept in mind and that is that as the outside temperature rises and the requirements become less, the temperature of the air or the quantity of the same, may be reduced, and in practice this is done so that the average amount of steam used for a season is much lower than the maximum requirements.

With a direct radiation installation the temperature of the steam cannot be lowered because generally exhaust or low pressure steam is used and hence the only way to cut down the quantity is to cut off certain parts of the radiating surface. This does not distribute the heat as well as before and therefore is not generally done. The method pursued is to open the windows and increase the radiation loss to a point where it will maintain a comfortable temperature and while the average outside temperature is higher than the maximum requirements the quantity of steam is not reduced materially.

Now as regards cost of installations. The design and manner of fan blast application is something that can always be worked out to good advantage though the method of application must necessarily depend upon the character of the building and its uses.

The ordinary industrial building presents the simplest of all problems. As a rule the per capita space for the workmen is large and the heating only is of paramount importance while ventilation is, in a way, incidental. As a rule ventilation can be secured by allowing the fan to draw its air from the building, thereby turning the air over and over and merely adding to the heat necessary to offset the leakage and radiation losses. To this end it is desirable that apparatus be placed as near to the center of the building as possible so the circulation is nearly uniform from all sides. Such a location very much simplifies the distributing arrangement and materially reduces the cost.

From the apparatus the air is discharged into underground or overhead pipes to get to its proper destination. It follows from the above, however, that the best results are obtained by discharging the heated air near the floor and outward so as to distribute it as nearly uniform as possible. It is usually most convenient to carry the piping overhead in the manner shown in Figure 2.

The average class of industrial buildings can very often be heated satisfactorily with a limited number of ducts by discharging the air at high velocity and thus compelling it to continue its direction of

movement for a considerable distance without the use of conducting pipes. This simple construction must be adapted to the character of the work carried on in the building, extra refinement in the manner of distribution being unnecessary where the workmen are actively employed. However, great care should be used in such a distribution not to have the air currents strike the occupants, as many systems have been condemned for this reason—no fault of the heating but of the distribution.

In other cases, when obstructions interfere, the air can be forced only a short distance and local distribution is necessary. This is common in the ordinary car paint shop. There the air is generally discharged downward toward the floor through pipes extending down between the cars. This not only heats the building but the time of drying is very materially decreased.

A similar problem is the case of a locomotive roundhouse where a blower system can be made to serve a double purpose. The general heating can be accomplished by discharging the air from overhead pipes toward the walls on either side or it may be utilized as a means of rapidly melting the snow and ice from the running gear of the engines by discharging all or a part of the warm air into the pits.

Industrial buildings two stories and higher can very often be equipped and satisfactorily heated without any distributing pipe connections by having vertical flues built in the walls from which the air is discharged toward the opposite side through openings from 8 to 10 feet above the floor line. These flues add but little to the cost of the building if the same are taken care of in the architectural design.

Whenever the flues can be built in the walls and the distributing ducts are of moderate extent the system will figure less in first cost than any other capable of attaining the same results.

From the standpoint of equivalent results a blower system of heating can be installed at no greater expense and with as much profit to the contractor as a system of direct radiation. It can be overloaded or can be operated as conditions require. In this way the steam consumed is minimum. If motors are used it might cost more to operate the fan but the saving in steam would more than offset this. There is the added advantage that ventilation is possible and where moisture is to be removed there is no comparison, as circulating air is the universal absorbing medium. In many cases, for no greater expense during the year, ventilation and heating can be produced for what heating formerly cost.

One feature of hot blast heating that is particularly apparent to us is that most contractors do not have the confidence in their ability to make the installation a success that they do with direct radiation. As a result, very often direct radiation is resorted to where a fan system would be more satisfactory and economical. We think this is due possibly to the feeling in general that a fan system is more or less guess work. Nothing is further from the truth, for the science of hot blast heating has been developed until absolute results can be guaranteed by the leading concerns.

We have a suggestion to make to the society in closing which we think worthy of your consideration.

From what we can learn we have no standards of heating or methods of determining heat losses. We have no standards of ventilation, or methods of testing installations. Frequently we see specifications for electric motors requiring that they comply with the requirements of the American Institute of Electric Engineers. Why should not the heating of a building be figured according to the standards of the A. S. of H. & V. E.? This would protect the public and improve the standing of the business and eliminate the class of guessers so detrimental to us all.

E. L. HOGAN, Chairman.
J. H. KASSA,
C. C. CHEYNEY,
Committee.

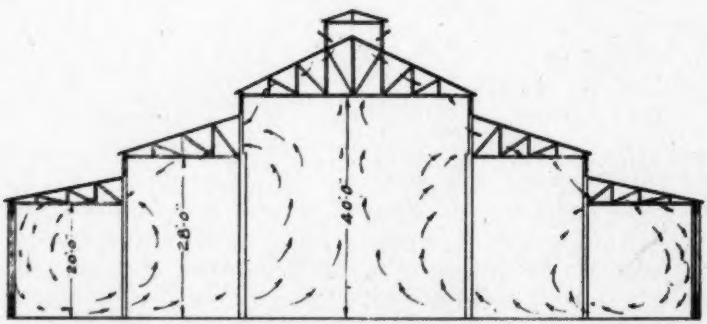


Fig. 1

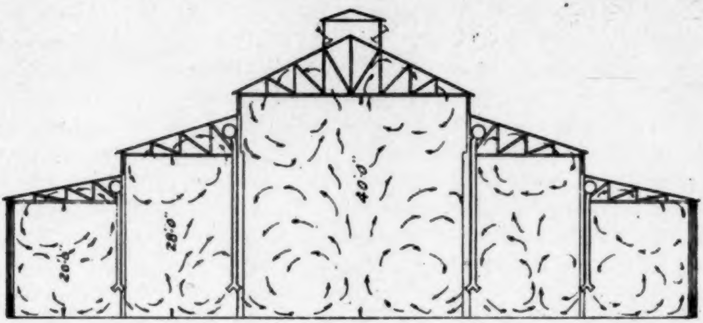


Fig. 2

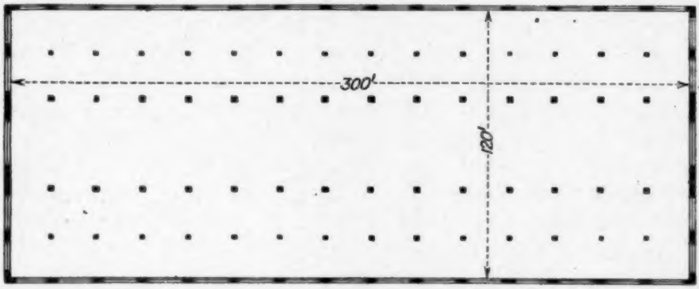


Fig. 3

Synopsis of paper

The paper gives floor plan and sections of a typical factory building of one story with a monitor extending the full length. It gives the wall and glass exposure and the heat loss in B.t.u. The apparatus described is that of a forced circulation warm air system through galvanized iron ducts.

The statement is made that such a plant would cost less to operate than would be the case with a direct steam plant in a similar building.

DISCUSSION

Mr. F. K. Chew: I suggest that this paper be given to the Committee on Standards to look into the advisability of having a standard for heating buildings in the same way that another committee has given standards for ventilation of buildings. Would this come up better under new business, or is now the

best time to take up the recommendations that Mr. Hogan's committee has made?

Mr. J. J. Blackmore: I would like to reply to Mr. Chew's suggestion that the paper brings up the question of cost of operation, claiming that indirect heating is cheaper than direct, as far as operating expense is concerned. Most of our members have but little data on the question of costs of operating plants. The President's paper gives some light on the subject, but the amount of data that we have is so small, it is impossible to derive much value from it. We should obtain data as to the cost of operating plants under all kinds of conditions and with different methods of heating.

President Lewis: This paper should really go to the Committee on Standards. I believe it will go to that committee. It it is your pleasure to consider discussion of the small paper which your President has submitted on operating costs, it can be taken up at this time or when we are discussing factory ventilation.

We need some standard unit as a basis by which we can accurately compare the cost of operating different plants, say a word that means this, "what it costs to supply one hundred thousand heat units." This would give us a basis of comparing cost of heating buildings that is more satisfactory than by cubic feet or by square feet of floor space.

Dr. Franklin: This would be expressed by mathematicians as "cost per hundred thousand heat unit years."

President Lewis: I agree that the only fair way to compare cost of heating buildings is by the cost per heat unit year, to borrow Dr. Franklin's expression. It is the only fair basis of comparison which should be adopted, rather than any attempt to compare such by square feet of floor space or by cubic feet of contents.

Mr. R. P. Bolton: An injustice has been done to the hot water circulating system, which for factory purposes, is meeting with a great deal of success, by reason of the ready control of the elements in two directions, namely, its quantity and temperature. In the operation of a hot water circulation system in an industrial building I found a situation which decidedly needed investigation by such a committee as that which we propose to have deal with this subject, because in both of these buildings a very large amount of radiation was supplied to a group of buildings, without any care as to the particular needs of the individual buildings. Where a number of buildings are heated as a unit the process must be productive of loss. In five buildings

of great magnitude the temperature is being controlled by one thermometer. Observations as to the cost of heating that group of buildings would mislead you, therefore, the question of separate division of such apparatus is of great importance. The difficulty in dealing with a subject of this kind is that it is so wide and varied, that the committee could not arrive in one year's time at any definite conclusions, hardly even as to the scope of what we want to find out. Climatic conditions are so varied, that it is only by confining ourselves to one phase and studying it for a long time, that we can decide the heating requirements in one neighborhood.

The study, however, is one that should be made, and I suggest as a method of working out some basis upon these points, that our members shall communicate their own observations made in different parts of the country, to this committee. I think we might join with other societies who are working along the same lines in endeavoring to secure information upon this matter.

There is more to be said on that subject by a great deal. In other words, there are some heating systems which are to-day installed at a capital expense, of which the use is very small, and it is questionable whether we get any commercial value out of the investment put into heating plants over and above a capacity to heat the building to 58 or 60 deg.

CCCLXXIII

AN EXPERIMENT WITH OZONE AS AN ADJUNCT TO ARTIFICIAL VENTILATION AT THE MT. SINAI HOSPITAL, NEW YORK CITY

BY A. M. FELDMAN

Another successful run during the summer of 1914 has been made of the cooling and ventilating plant in the small ward of the Mt. Sinai Hospital, New York. This plant was installed for the purpose of treating gastro-enteritis diseases of children. Although the success with which it was used in the summer of 1913 directly following its installation, was highly satisfactory, its usefulness was considerably augmented by the addition of an ozone machine the past summer.

The plant, as it has been described in a paper presented by the writer before this Society at its annual meeting of 1914, was designed to deliver a sufficient quantity of fresh, cleaned and cooled air, for an average of three small children. This equals about 250 cu. ft. per minute. The vitiated air is allowed to escape through two open window transoms.

Notwithstanding the quantity of fresh air supplied there always has been a decidedly unpleasant odor due to the character of the periodical discharges by the children under treatment. The writer felt that an attempt to eliminate this odor by means of an augmented delivery of fresh air would have been prejudicial to the safety of the children and nurse in charge on account of draughts and aside from this objection, the cooling machine would have proven inadequate for cooling the increased amount of air.

The writer, therefore, thought it would be a good opportunity to test the efficacy of the claim being made for ozone as a neutralizer of odors. Through the kindness of the Hudson Ozone Machine Co., which volunteered to furnish the writer with some of its small machines, the experiment to which reference is made, was accomplished.

The machine was placed at the end of the room farthest away from the fresh air register—near the window. The small fan in connection with this type of machine was removed to prevent the causing of draught by the violent movement of the cool air.

It is gratifying to the writer to be able to report that the experiment proved a success. That the use of ozone has neutralized the odor above referred to, has been attested to by the nurse in charge, and by the attending and visiting physicians. It was especially marked when, on a few occasions, the ozone machine got out of order. The nurse was then very anxious to get it back into use, as she had been rendered more strongly susceptible to the odor by its very absence.

This single personal experience has satisfied the writer of the fact that there is merit in the claim for ozone as an adjunct to artificial ventilation in cases where objectionable odors are being constantly generated.

Synopsis of paper

In connection with ventilation it describes the use of ozone to destroy the odor emanating from the children under treatment at Mt. Sinai Hospital, and the paper tells us that it was successful.

DISCUSSION

Mr. Thomas Barwick: What was the quantity of ozone used in this particular room? This, of course, will make a difference as to whether the ozone was greater than was absolutely necessary or whether it was proportionate to the amount of fresh air supplied to the room. I have found in some cases where ozone has been used that there is a considerable smell from the ozone that made it objectionable. I had a case where four machines were used. It may be I had more than was absolutely necessary. The question arises how many mm. of ozone are to be provided for a given quantity of air supplied?

Mr. A. M. Feldman: This was not intended to be a test of the ozone machine, therefore, I did not mention quantity; as I did not go into measurements. The Hudson Company, who provided the machine, designates it as size for 250 cubic feet of air. The size of the room was about 12 x 9 x 13 ft.

Dr. M. W. Franklin: This experiment of Mr. Feldman is not unique. Dr. Van der Bogert, connected with the Ellis Hospital, in

Schenectady, has made similar experiments, and I may sometime be able to induce him to report his results. It seems to be very well demonstrated that ozone is capable of overcoming odors without interfering with the health of the occupants of a room, if used properly. The question of how much can be used is debatable. The question that has been asked is whether the ozone produces olfactory compensation of the odors, or destroys them. In the last two years I have written two papers, which I consider absolutely convincing. The manner in which the experiments have been conducted demonstrate beyond the possibility of a doubt that the odors are destroyed. I have taken certain known definite chemical compounds and have arranged the experiments so that this could be demonstrated. In this case it could not be said that the ozone causes olfactory compensation of the odor. In the case of organic substances where the odor disappears, it is reasonable to suppose, even in the absence of exact chemical analysis, that the substance has been destroyed. Furthermore, the methods of experiments have been such that the ozone could not have been present in the final test.

The application of ozone to ventilation is limited. It certainly will destroy odors and in so far as they are objectionable, it can be employed with assurance of success. Ozone, properly used, is unquestionably the proper medium and this is being demonstrated further every day.

Mr. Thomas Barwick: The idea is to get the quantity that is necessary for one thousand cubic feet of air or one hundred feet, to destroy a certain odor. We laymen do not know how many mm. of ozone are required for so many thousand feet of air, and if Dr. Franklin can give us some sort of proportion, so we can base our calculations, we will be placed in a very much better position than we are at present.

Dr. M. W. Franklin: I have seen fruit maintained for four weeks with ozone. There are many possibilities for ozone, not only for destroying odors, but for an adjunct in refrigeration and in other things.

Mr. A. M. Feldman: What is your opinion whether the odor is only neutralized or destroyed?

Dr. M. W. Franklin: If you will read my first paper, a copy of which I will send you, one of the subjects on which I experimented was human faeces. We have destroyed odor in a way that I fail absolutely to see wherein any error could be, and I have submitted

these results to the best qualified experts I know of for their opinions, and they have all agreed with my conclusions. There can be no question as to whether the odors are destroyed or merely compensated. The fact remains that we have every proof in the world, including, in many cases, chemical analysis, that the odors are absolutely destroyed, and not covered by ozone.

CCCLXXIV

A STUDY OF HEATING AND VENTILATING CONDITIONS IN A LARGE OFFICE BUILDING

C.—E. A. WINSLOW* AND G. F. MAGLOTT

The progress of the art of heating and ventilation has been seriously retarded by the gap which unfortunately often exists between design and operation. Excellently planned systems may fail on account of changes in conditions of occupation or carelessness in upkeep and management; while, on the other hand, operation sometimes reveals shortcomings in design which should be instructive in the planning of future installations. Careful studies of the actual results obtained in practice from the operation of heating and ventilating plants are none too common. The results of such a study of a large business office building in New York City may therefore be of some interest. The authors had the advantage of the advice of Mr. D. D. Kimball as consultant in the course of the investigation.

THE SYSTEM OF TEMPERATURE CONTROL

The building is heated in the main by direct steam radiation although certain rooms on the lower floors are in part indirectly heated by plenum air supply. The steam in the heating coils is maintained under half a pound pressure against a one inch vacuum.

The radiators in certain sections are hand controlled, in other sections thermostatically controlled. Of the workrooms examined by our inspectors, 65 had hand-control and 46 thermostatic control; of the offices 39 had thermostatic control and 82, hand-control. The thermostats in use are of an old type and are in a state of general disrepair. Of 213 thermostats individually examined, 110 or 52 per cent. were not controlling their radiators. A considerable

*Curator of Public Health, American Museum of Natural History; Chairman, New York State Commission on Ventilation.

part of the trouble was obviously due to interference with the instruments on the part of clerks who, dissatisfied with existing temperature conditions, have attempted to set them right without understanding the mechanism with which they were tampering.

TEMPERATURE AND HUMIDITY CONDITIONS IN OFFICES AND WORKROOMS

During the period between January 25th and May 17, 1914, our inspectors made 2,045 determinations of temperature and humidity by the use of the sling psychrometer in practically all the offices and workrooms occupied by the company.

The humidity results were not particularly significant, merely showing that as usual the indoor humidity varied directly with the outdoor temperature. During most of February and the first half of March the humidity in the building was under 30 per cent. of saturation, in the last half of March and in April it fluctuated between 30 and 40 per cent. and in May it rose above 40 per cent.

The temperature data were of course of much greater practical importance. The average weekly temperature for the building never fell below 69 degrees during the whole period studied. Out of the 16 weeks covered by our investigation, four showed averages between 69 degrees and 70 degrees, ten averages between 70 degrees and 72 degrees, one an average of 73.5 degrees and one an average of 75 degrees.

Fifty-eight per cent. of the individual rooms studied showed a temperature of 71 degrees or less; thirty-four per cent. were between 72 degrees and 75 degrees; nine per cent. were between 76 degrees and 81 degrees. The distribution of observations is shown graphically in Fig. I, the hand controlled and thermostatically controlled rooms being plotted separately. For comparison we have included the distribution of 143 temperature records made by Professor Baskerville and the senior author in a well managed public school, School 33, the Bronx, New York City.

The curves in Fig. I bring out two points; first, that the Office Building as a whole is greatly overheated as compared with such a constant low temperature as is maintained in School 33 (in which ninety-seven per cent. of all records were under 72 degrees); and second, that overheating is even slightly greater in the thermostatically controlled than in the hand controlled rooms. The moral is that overheating is likely to be general either where hand control alone is relied upon or where a thermostatic system has been allowed to fall into disrepair.

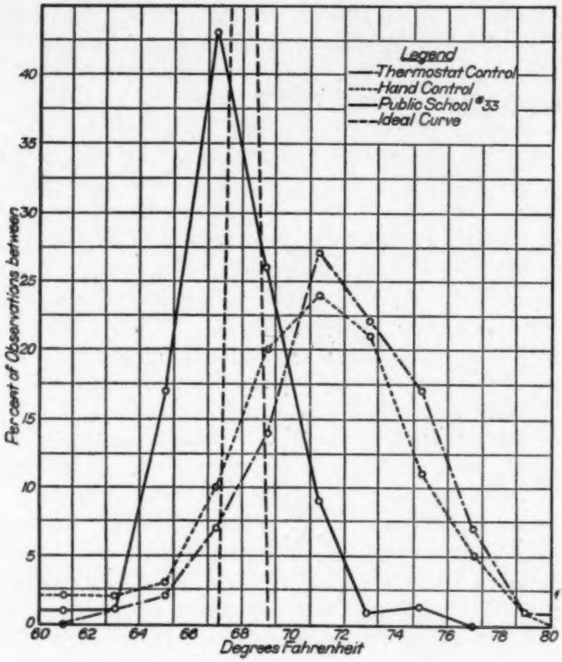


Fig. I.

In addition to the sling psychrometer determinations we obtained continuous records of temperature by means of Tycos thermographs in twenty-five different rooms, usually for a two weeks' period in each room. The record for one of the most overheated rooms in the building is reproduced in Fig. II to show how bad conditions may become. This particular room had three thermostats, all in disrepair; and for a period of twenty-one days, between February 28th and March 24th, the temperature never once fell as low as 70 degrees during working hours, and was usually in the neighborhood of 80 degrees.

THE SERIOUSNESS OF OVERHEATING

It is perhaps hardly necessary before this Society to dwell upon the importance of such conditions of overheating as those recorded. It has again and again been shown that the most serious of the effects experienced as a result of bad air conditions are due, not to chemical poisons but to high temperature. The recent studies of the New York State Commission on Ventilation furnish striking

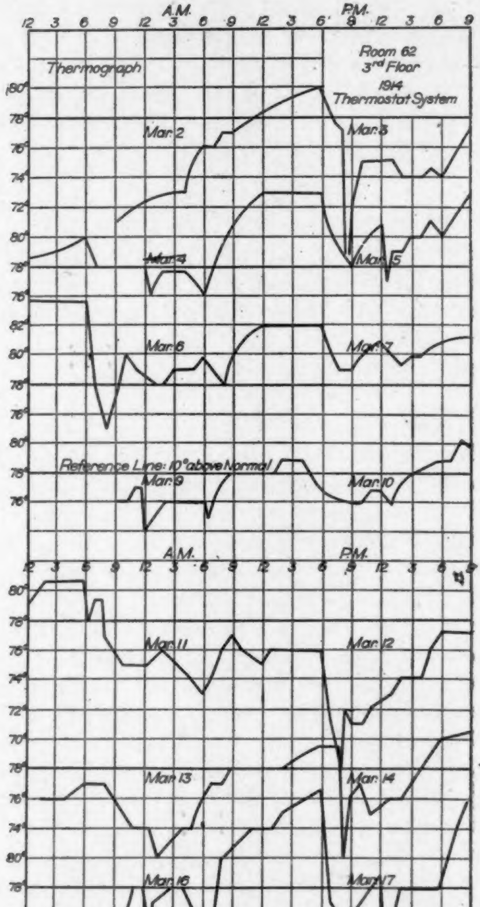

A.M.
P.M.
Thermograph
Room 62
3rd Floor
1914
Thermostat System
Mar. 2
Mar. 3
Mar. 4
Mar. 5
Mar. 6
Mar. 7
Reference Line: 10° above Normal
Mar. 9
Mar. 10
Mar. 11
Mar. 12
Mar. 13
Mar. 14
Mar. 16
Mar. 17
Mar. 18
Mar. 19

Fig. 11.

evidence of the bad effects of high temperatures upon the heat-regulating system of the body as shown by rise in body temperature and fall in blood pressure and by the decrease in mental and physical work done in overheated rooms. There can be no doubt that the clerks exposed to such conditions as those observed are injured in health and hampered in efficiency thereby.

THE VENTILATING PLANT

Considerable areas in the sub-basement and basement and a few company offices and workrooms on the first, second and third floors are equipped with plenum air supply. The rest of the building has either an exhaust system alone or no artificial ventilation at all. About 44 per cent. of the workroom area comes under the latter head.

It may be generally assumed that when an office or workroom is so crowded that the floor area is less than 200 square feet per capita, efficient artificial ventilation will be necessary. Of all the workrooms occupied by the company we found that 90 per cent. have less than 200 square feet per capita, 78 per cent. less than 100 square feet and 31 per cent less than 50 square feet. The provision of only 56 per cent. of the workroom area with any artificial ventilation at all is therefore the first serious defect indicated.

The 56 per cent. of the workroom area which does have artificial ventilation depends almost wholly upon the exhaust system. This system consists of six separate shafts, one of which is much larger than the others. Each shaft is of graduated cross sections carried up through the building to a fan room at the roof. Two of the shafts (IV and VI) are lined with sheet metal, the others are of plaster on metal lath carried on light steel shapes. All the fans are of the propeller type operated by direct connected electric motors, controlled by rheostats. All the fans were found to be operated at a reduced speed in accordance with orders of the chief electrician.

The general engineering data obtained in regard to the operation of these ducts are indicated in Table I. We have estimated the probable population on each shaft in 1925 as indicated by the past growth of the business and have tabulated the work done by the fans as at present operated and when operated under our direction at full speed.

Our results indicate that with the present population all the shafts but II are exhausting over 30 cubic feet per capita and may be considered satisfactory. Shaft III, however, is close to the limit and Shaft II is already inadequate.

TABLE I

SHAFT AND FAN CAPACITIES

No. Shaft	Population		ha ft	Present Speed			Full Speed			Q Per Capita		Rated or Theoretical	Rated H. P.	Diam.	Capacity of Shaft*	
	Present	1925	Area Top sq. ft.	R P M	Vel. Ft. per min.	Q	R P M	Vel. Ft. per min.	Q	Present	1925	Q		Fan inches	Total Q	Q Per Capita 1925
I	437	700	27.3	368	475	12950	438	570	15500	36	22	20000	5	48	21800	31
II	533	870	16.2	334	500	8100	450	740	12000	23	14	25000	5	48	13000	15
III	411	670	16.5	240	485	8000	418	795	13100	32	20	15000	5	48	13200	20
IV	483	790	18.0	225	1100	20000	268	1310	23600	49	30	37000	6¼	72	14400	18
V	25	40	11.7	476	410	4800	600	490	5700	228	142	5000	1¼	30	9400	235
VI	1100	1810	70.5	112	733	51700	160	970	68300	62	38	75000	11½	120	56200	31

*Figures based on a velocity of 800 ft. per minute as limited by type of fan. With a different type of fan the capacity of the shaft would be doubled.

ROOM REGISTER OPENINGS AND PER CAPITA AIR SUPPLY

The most serious defect of the ventilating system we found to lie in the generally inadequate room register openings which in many cases prevent the main ducts from effecting a change in the rooms which need it most. In order to remove the standard air quantity of 30 cubic feet (per minute) per capita with the register velocities practically attainable it is necessary to have a register area of about .1 square feet per capita. A detailed study of 74 workrooms showed that in 23 cases or 31 per cent. the register area was less than .075 square feet per capita, in 12 cases or 16 per cent. between .075 and .125 square feet, in 16 cases or 22 per cent. between .125 and .200 square feet, in 8 cases or 11 per cent. between .2 and .3 square feet in 13 cases or 18 per cent. between .3 and 1.0 square feet and in 2 cases or 3 per cent., over 1.0 square feet. This means that the relation of register area to room population under present conditions of occupancy has become quite haphazard, and that 31 per cent. of the workrooms have less than the proper allowance of register area while others have far too much.

Determination of the actual flow of air at the exhaust registers was made with the anemometer on 148 different occasions in one or another of the fan-ventilated workrooms, practically all of them being covered at least once. Ten rooms or 7 per cent. showed a negative pressure (the supposed exhaust acting as a supply), 47 or 32 per cent. showed less than 10 cubic feet per capita, 32 or 22 per cent. between 10 and 20 cubic feet per capita, 18 or 12 per cent. between 20 and 30 cubic feet, 11 or 7 per cent. between 30 and 40 cubic feet and 30 or 21 per cent. over 40 cubic feet per capita. Thus in 89 cases out of 148 or 61 per cent. the air exhaust was quite inadequate, being less than 20 cubic feet per capita, in 29 cases or 19 per cent. it was between 20 and 40 cubic feet, a reasonable amount, while in 30 cases or 22 per cent. it was over 40 cubic feet or far in excess of necessary requirements.

It was interesting to note that by far the worst conditions were found upon the upper floors. In the lower stories the exhaust was usually adequate while in the upper stories the amount of air drawn out progressively decreased. On two of the shafts the suction above the seventh and eighth floors respectively was either negligible or was replaced by a positive outflow of air into the room, vitiated air from the lower workrooms being forced out into the upper ones (Fig. III).

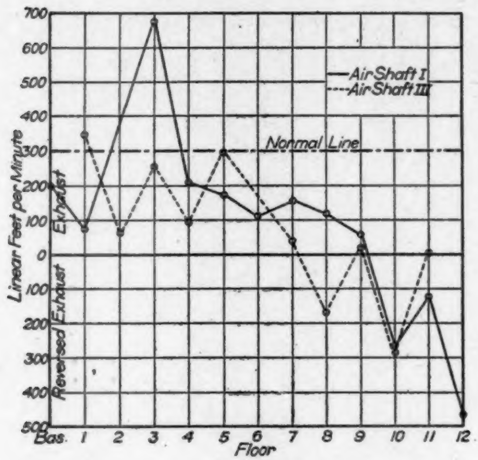


Fig. III.

DETERMINATION OF CARBON DIOXIDE

In order to measure the actual extent of air change within the offices and workrooms we made 228 determinations of carbon dioxide using the standard Petterson-Palmquist apparatus. Of the 228 records 58 per cent. were below 9 parts per 10,000, 32 per cent. between 9 and 12 parts, and 12 per cent. over 12 parts, indicating that air change was inadequate in over 40 per cent. of the cases studied.

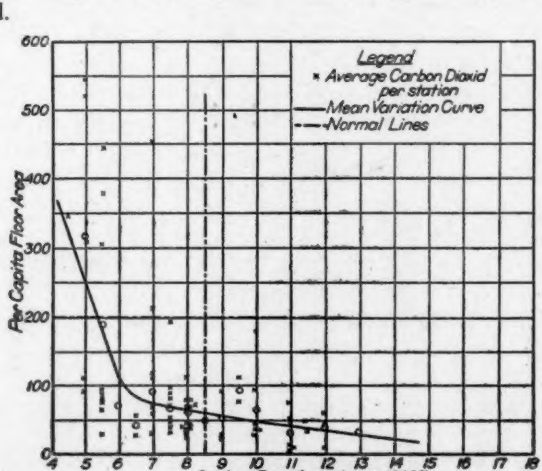


Fig. IV.

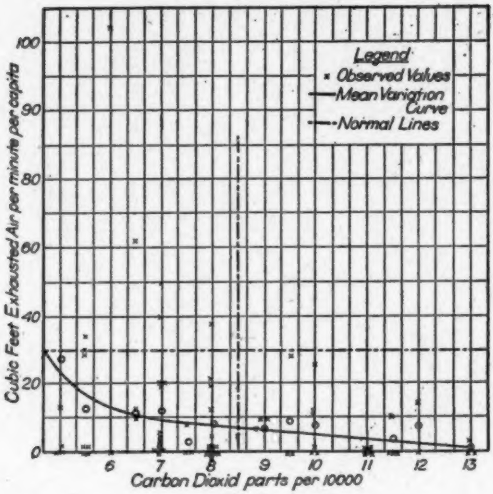


Fig. V.

As shown in Fig. IV the worst results were of course in the worst crowded rooms. It will be noted that taking 8 parts of carbon dioxide as a reasonable limit only one excessive value was recorded in a room with over 200 square feet of floor space per capita.

Fig. V shows the relation between carbon dioxide and air movement at the register openings and it will be noted that only one high value was obtained where the air exhaust was over 30 cubic feet per capita. Mean carbon dioxide values increased progressively in passing upward from floor to floor.

The actual air conditions in the workrooms were of course controlled not only by the exhaust system but also by the extent to which windows were kept open. In Fig. VI the observed carbon dioxide values are plotted against the area of open window space in the respective workrooms at the time expressed in square feet per capita. The relation is seen to be a very close one. With an open window area of over .5 square foot per capita, not a single high carbon dioxide value was recorded and only six values were obtained with an open window area equal to .25 square foot.

Table II shows the carbon dioxide values obtained, classified according to the condition of the respective workrooms in regard to both exhaust ventilation and extent of window openings. The table indicates clearly that with an adequate exhaust and open windows conditions are always excellent, that with either adequate exhaust or open windows intermediate conditions are obtained

while results are generally bad when neither natural nor artificial ventilation is available.

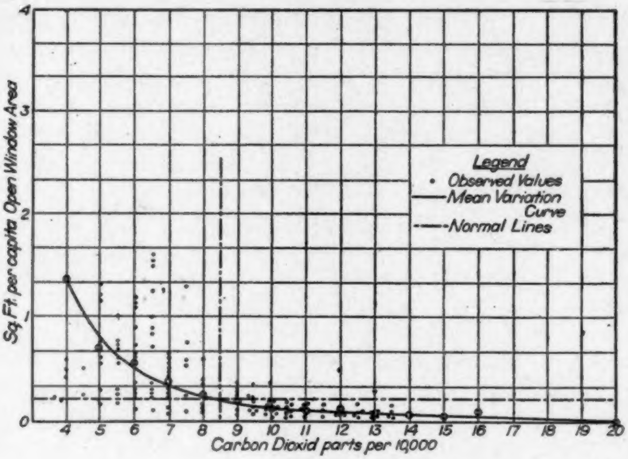


Fig. VI.

TABLE II

Relation of Artificial and Natural Ventilation to Air Conditions

Carbon Dioxide, Parts per 10,000.	4	5	6	7	8	9	10	11	12	13	14	15	16
Rooms with exhaust over 30 cu. ft., window openings over .25 sq. ft. p. c.	5	9	8	6	2								
Rooms with exhaust over 30 cu. ft., window openings under .25 sq. ft. p. c.......		1	1	2	1	1							
Rooms with exhaust under 30 cu. ft., window openings over .25 sq. ft. p. c.......			12	14	11	2	2		1	1			
Rooms with exhaust under 30 cu. ft., window openings under .25 sq. ft. p. c...	1	1	3	6	6	4	13	11	5	6	2		1

SPECIAL PROBLEMS OF LUNCH ROOMS AND AUDITORIA

On the two upper floors are two large lunch rooms and several small lunchrooms for the clerical force and a large auditorium.

The register areas serving the lunchrooms were found to be so inadequate as to be practically negligible and the flow of air through them was slight or even reversed in some cases. Of twenty-three temperature records during lunch hours four were between 72 degrees and 73 degrees, seven between 74 degrees and 75 degrees, six between 76 degrees and 77 degrees and six between 78 degrees and 79 degrees. Of twenty-three carbon dioxide observations one was 8 parts, eight 10 parts, two 11 parts, three 12 parts, three 13 parts, two 14 parts, three 15 parts and one 20 parts. In the auditorium conditions were even worse. This room has a fan and supply and exhaust ducts adequate to all demands but the motor is too small to operate the fan at full speed and the register openings for both incoming and outgoing air are located far above the heads of the occupants. The air enters at the ceiling about twenty-one feet above the floor and leaves about sixteen feet above the floor so that the occupants are not materially benefited. Observations made on five different occasions when the room was in use gave temperature averages of 68, 75, 87, 87 and 87 degrees and carbon dioxide values of 12, 10, 8, 20 and 13 parts, respectively.

GENERAL CONCLUSIONS AS TO VENTILATION

According to the standard of 200 square feet of floor area per capita, which we consider a reasonable one, 90 per cent. of the workrooms in this building are so crowded as to be in need of artificial ventilation. Out of 118 workrooms studied 41 or 35 per cent had no artificial ventilation at all. Of those which were provided with exhaust ventilation, the per capita air movement was inadequate in 61 per cent. of the cases studied. Adding to the 35 per cent. of the rooms which have no artificial ventilation, 61 per cent. of the other 65 per cent. or 39 per cent. of the total we find that 74 per cent. of all workrooms are either not ventilated or ventilated insufficiently. Carbon dioxide determinations confirm this general conclusion indicating poor air conditions in 42 per cent. of the rooms studied. In certain cases, of course, window ventilation made good the lack of artificial ventilation.

Only by either (a) artificial ventilation exhausting at least 30 cubic feet per minute per capita or (b) an area of open windows equal to at least .25 square foot per capita could reasonably low carbon dioxide values be obtained. The usefulness of window ventilation is greatly limited by local and climatic conditions. In warm weather there is no force to drive air in through the windows; in cold weather drafts are objectionable and windows are closed; and

at all times the noise and dust of the city streets constitute serious objections. The inadequate artificial ventilation of the building is made obvious to the senses in many of the workrooms by the stale, stuffy odor characteristic of foul air.

In general, then, our investigation showed that both the heating and the ventilating systems had been allowed to fall into such disrepair and to become so ill adjusted to present needs as to fall far short of realizing the purposes for which they were designed. It is just such conditions as these which constantly bring discredit upon the art of heating and ventilation, conditions which can only be brought to light by a comprehensive engineering and sanitary study of actual operation.

Synopsis of paper

Describes the system of temperature control.

The temperature and humidity conditions in the offices and work rooms.

Discusses the seriousness of overheating. Describes the ventilating plant, its capacity, condition and efficiency at the time of the examination. Describes the room register openings and the per capita air supply.

Shows several charts of determinations of CO_2 content. It also offers some conclusions as to standards.

DISCUSSION

Mr. F. K. Davis: Prof. Winslow states that during February and first half of March the humidity of the building was under 30 deg. About five years ago, I had occasion to take some humidity readings in an office building in Baltimore. I found that in the corridors and generally throughout that building that when temperature outside was at or below 30 degrees, their highest percentage of humidity was 21, and a fair average, I should say, was about 19. Those readings were taken with temperatures outside of 14 degrees up to probably 40 degrees, and the building was heated throughout with direct radiation.

In speaking of the dust in connection with open windows, I personally knew of a case where they had an old gravity indirect ventilating system with stacks, and the heating contractor told me he got eight cart loads of stable manure from the indirect casings at a time when they were overhauling and putting this plant in order. The point Prof. Winslow brought out covers the

subject closely in practice. It is a step in the right direction, and I am glad to see that Prof. Winslow has taken the stand that ventilation by open windows is more or less a mistake.

Mr. H. M. Hart: I think that something ought to be done to get the co-operation of the operating engineers through their local societies, to take more interest in the efficiency of ventilating systems. It seems to be the tendency on the part of the engineers to make a good showing at the end of the year on the cost of operation only, rather than from the results he gives the occupants, and I believe in 90 per cent. of the installations of good ventilating equipment the chief engineers instruct their operators to run fans at a reduced speed in order to save coal and current, in order to make a good showing and make the owner think that he is a good engineer and a good investment for him.

It seems to me that if this was put up in the proper manner to the Stationary Engineers' Societies, by members of this Society, some good could be brought about. If we could arouse their pride in the fact that they are running a system as designed and getting good results, rather than saving in dollars and cents in the coal pile; and get the owner to appreciate what a well designed system is, we will be taking a step in the right direction.

Mr. Baldwin: There is a hospital within one mile of us (a great charity) that was put up by a well known gentleman and his wife in New York (rich people), and which bears their name. They made arrangements with the adjoining building to take steam from them from which they receive a revenue. The original fans in that building were run at two hundred revolutions per minute to accomplish normal results. If you will go there now, you will find those fans are just creeping around, less than 50 R.P.M. It is done to show a saving in coal.

I am taking the air from the top of the house in a large city hospital at this time. When this particular building was planned, the first question asked was: "Where will we take the fresh air from?" I was the engineer. We built a shaft to the top of the building and took the air 150 feet above the ground, and thought we were going to have splendid results. We erred. There were abattoirs near the river front, and we got their smoke, and the air that was taken in was carrying dust from the neighboring chimneys.

When an extension was built, after a few years, we took the air from about 75 feet from the ground, and the conditions were not much better.

When the third installation was put up, we built a pagoda in the middle of a vacant grass plot, and the results were very much better, though the air was taken almost at the street level. These incidents are worth remembering.

Pres. Lewis: There are two ways to overcome that, that seem to be practical. One is the method used in Chicago. Each inspector is constantly visiting all of the plants in his district, seeing that they are operated. A suggestion was made at the time the department was organized that a meter be devised which could be put on each fan which would show the number of hours it has been operated between the times the inspector visits the plant.

Another way which I personally adopt is to install such a plant that they cannot live in the building unless they run the fan; that is to make the heating and ventilating combined, so that it is necessary to run the fan to maintain a living temperature in the rooms.

Mr. J. H. Davis: Mention has been made of Inspectors and a fear expressed that we are liable to have inspectors who are merely political appointees, without regard to fitness. This is not always so. In Chicago these inspectors are under the control of the Health Department, and have to pass a special examination before they are appointed. My experience with these inspectors has been that they are all trying to learn as much as possible of the art of heating and ventilation and its application to the buildings under their supervision. The result is we have in Chicago inspectors who are thorough in their work, honest in their reports, and it is being demonstrated that we are getting admirable ventilation in our theatres, halls, etc., in accordance with the Ventilation Ordinance.

CCCLXXV

THE VENTILATION OF SLEEPING CARS: COMPARATIVE TESTS OF VARIOUS TYPES OF EXHAUST VENTILATORS

THOMAS R. CROWDER, M.D., CHICAGO

Perhaps the time has not yet come when we can say that fresh air is thoroughly understood. But we can certainly say with safety that popular conceptions concerning it are very erroneous. It is still commonly held that out-breathed air contains a volatile poison, and on this erroneous assumption have been developed the theories which have controlled the practice of ventilation almost up to the present time. There can of course be no doubt that the air confined in crowded places may become harmful through changes brought about by the people using it, and that it needs to be frequently renewed. The necessity for ventilation is beyond question; but the necessity does not arise from a poisoning of the air by the products of respiration.

I.

The air which surrounds the body has two principal functions: a chemical and a physical. It oxygenates the blood and it removes the body heat. For the performance of its chemical function it must contain a sufficient amount of oxygen to keep the hemoglobin saturated and be free from poisonous gases; for the performance of its physical function it must be cool enough to absorb the heat of the body, dry enough to take up moisture from the skin, and have motion enough to carry away the aerial envelope to which this heat and moisture are transmitted. If the air of the room is not renewed its oxygen is gradually consumed and it becomes laden with heat and moisture from the bodies of the occupants. In this way it may finally become unable to perform either of its principal functions. A constant supply of fresh air is therefore necessary. But careful

experiment has demonstrated that under all ordinary circumstances the fault develops on the physical side so far in advance of the chemical that the latter may be practically left out of consideration. Relatively small amounts of fresh air will always supply the chemical needs of the body; large amounts may be necessary to supply the physical demands. Granted that the small amount of air necessary for the demands of respiration is supplied, the control of its physical properties becomes the great problem of ventilation; and of these physical properties temperature is vastly the most important. The success of ventilation depends far more on supplying conditions suited to the outside of the body than to the inside of the lungs.

The above is a very brief statement of the fundamental principles of ventilation as I understand them. Time will not allow me to discuss in detail the evidence on which these principles are based, and with which I have no doubt you are already familiar, but I desire to emphasize a few points in connection with them.

Though a vast amount of experimentation has been devoted to proving that the expired air contains a volatile toxin this hypothetical substance has constantly failed of demonstration; and such negative evidence has been now accumulated as seems to clearly prove that there is none—at least that it is not present in sufficient quantity to be of any practical importance.

Recent developments in experimental physiology have also demonstrated that harm does not come through rebreathing the exhaled breath. With a proper understanding of the process of respiration it was never a reasonable supposition that such rebreathing should be harmful; for breathing is essentially only a frequently repeated slight dilution of the alveolar air, which always remains the chemical equivalent of expired air. At each breath we take back into the lungs the expired air which is lodged in the nose and larger bronchi—the so-called "dead-space" air. This dead-space equals about one-third of the volume of a quiet inspiration. To all intents and purposes it contains air which has passed out of the lungs and is reinspired. It is seen, therefore, that the air of the lungs is not only in itself very impure—that it never even remotely approaches pure air—but that pure air is never allowed to enter the air cells of the lungs from without.

Reinspiration as brought about in the way described is more than a physiological accident. It is useful and necessary. It makes of breathing a continuous instead of an intermittent process and so provides for a constant supply of oxygen which is necessary to the tissues.

The nerve center which controls respiration depends for its stimulation on carbon dioxid dissolved in the blood. If the carbon dioxid falls too low, stimulation does not take place until the proper proportion is reaccumulated. Reinspiration of the dead-space air prevents the too rapid escape of CO_2 from the blood; for when the blood passes through the lungs it loses its dissolved carbon dioxid to the air in them until the pressure of this gas in the blood has fallen as low as is its concentration on the outer side of the membrane separating the blood stream from the air cells. When the pressure of CO_2 in the blood falls below about five per cent. of an atmosphere, breathing ceases until this level is restored, since the respiratory center is not stimulated by less.

Contrary to common opinion the want of oxygen has nothing to do with stimulating the respiratory center, nor does a deficiency of oxygen in the blood, independently of the presence of CO_2, make itself known to any nerve center which will reflexly supply the demand. Oxygen starvation may coexist with the absence of breathing if CO_2 is first removed from the blood. Alveolar air normally contains about 16 per cent. of oxygen, and the red blood-cells leave the lungs practically saturated with it. The amount taken up on their next trip through the lungs depends on how much they have given up to the tissues in the meantime, not on how much is available to their use. The normal 16 per cent. of oxygen in the alveolar air is automatically maintained by the action of CO_2 on the respiratory center, but on account of the chemical affinity of the hemoglobin for oxygen the blood-cells may still take practically their full capacity when it is reduced to 12 per cent. or less in the alveolar air. Thus a large excess of oxygen is constantly maintained in the air of the lungs. And while it is one of the chief functions of respiration to supply oxygen to the body, neither a surplus nor a deficiency of it, unless the alteration is extreme, have any effect on the respiratory movements. Breathing will not be lessened nor more oxygen taken up because more of it is supplied to the lungs; nor will the oxidation processes in the body be affected in any way, unless other influences are simultaneously brought into play.

It is thus seen that the maintenance of a high concentration of the chief product of expired air—carbon dioxid—in the air of the lungs is our chief safeguard against the want of oxygen by the body tissues, and that rebreathing the dead-space air is an important measure by which this high concentration is maintained.

Besides the necessary reinspiration of the dead-space air, it is also known that one usually rebreathes a part of the breath entirely expelled from his body during the preceding expiration. In another

place I have presented the experimental evidence on which this statement rests. When standing alone in a well-ventilated room a person will rebreathe one or two per cent. of the air he has just exhaled, when lying in bed he will rebreathe from one or two to six or more per cent., depending on his position; and even in the open, if there is a shield to break the breeze, a small proportion is taken back with nearly every breath. To rebreathe one's own air is thus seen to be a natural and unavoidable occurrence, while to breathe some of the air exhaled by others is the common lot of men that live together in close proximity. It has been done through the ages, and with it have been developed the protecting devices necessary to give us a wide margin of safety.

For not only is the depth of breathing lessened when the CO_2 of the alveolar air drops below about five per cent. of an atmosphere, but it is increased if for any reason the proportion rises above five per cent. An increased concentration of CO_2 in the blood causes an excessive stimulation of the respiratory center, just as a lowered concentration causes a decreased stimulation of the same center. When one breathes in a little excess of his own expired air the alveolar CO_2 is not quite reduced to its normal five per cent., and the blood flowing on to the brain with a slightly exaggerated proportion of this gas causes the inspiration to be a little deeper; if the air breathed in contains a little less than usual of his expired products, the inspiration is a little shallower for an opposite reason. By reason of the nice adjustment of the reflexes controlling the depth of breathing, the concentration of CO_2 in the lungs is thus kept at its normal uniform level regardless of whether we rebreathe much or little expired air. The balance is maintained automatically and without our consciousness. And the limits of this automatic regulation of the volume of air inspired go far beyond the limit of the necessity for change placed upon it by any respiratory contamination that is ever likely to be found in the air of even the most crowded rooms.

But if the accumulated evidence of physiologic studies has taken from us the old basis of chemical purity as a guide to ventilation standards, it has at the same time pointed the way to a new and more logical basis existing in the physical properties of the air. It has demonstrated that the discomfort and physiologic disturbance experienced in badly ventilated rooms is due to derangements of the vasomotor mechanism brought about by the thermic influences of the air in contact with the surface of the body. They are due to the heat, the humidity, and the windlessness of the air which render it incapable of cooling the skin at the normal and necessary rate.

An ordinary adult will produce—and must be relieved of—enough heat in the course of an hour to raise the temperature of a thousand cubic feet of air by fifteen or twenty degrees Fahrenheit. The air with which the body is either directly or indirectly in contact must take away this heat. Herein lies the basic equation of the ventilator's problem. When many people are crowded together in a small space they very soon overheat the air, and its depressing effect is added to by its stillness and by the moisture exhaled with the breath. More air and cooler air must be supplied; fresh air becomes an imperative need. But here I would emphasize that the quality we generally recognize as "freshness" does not depend on richness in oxygen, nor on the absence of CO_2 and organic poison, but on the ability of the air to remove the body heat. Fresh air is air that will cool the body more rapidly. Failing in this, no property, either physical or chemical, will make it "fresh." Coolness of the air is more important than its content, and agitation is of greater significance than its purity. It follows that the impulsion of hot air into a room is the most objectionable of all systems of ventilation, while cool air and radiant heat approach the ideal.

But changing our scientific basis for ventilation does not necessarily mean that we must entirely desert all quantitative standards in actual practice. We must not expect the body to transmit its hundred calories per hour to a hundred cubic feet of air, though a hundred cubic feet may fully supply the demands of respiration. For the body at rest and with ordinary indoor clothing, there are sharp limitations to the physical conditions that will maintain the sense of well being. Within a range of temperature compatible with comfort it will require something like 2000 cubic feet per hour to absorb the heat of an ordinary adult, unless the heat transmitted to the air is rapidly abstracted from it. Curiously enough this figure corresponds closely to that arrived at long ago as the air supply necessary to maintain the requisite degree of chemical purity. The suggestion is forced upon us that the old standards were really, though not consciously, based on thermic considerations after all, for CO_2 is a guide to quantitative interchange as well as to chemical purity. But on the old basis, more air was the universal remedy for ventilation troubles—more air in order to lessen respiratory contamination—and it often failed to effect a cure. On the new basis, cooler air, or dryer air, or more motion—all of which facilitate the transfer of body heat, and none of which necessitates an increased supply—are the measures indicated; and through a proper combination of these remedies relief should be obtained.

II.

It is now seven or eight years since I began to study the ventilation of sleeping cars and to attempt a solution of the vexed question concerning the best methods for producing comfortable and hygienic conditions in them. In common with most people I then believed these cars had a very inadequate air supply. As we now understand the subject, it must be admitted that the cars were poorly ventilated, and that the air supply was sometimes small, but faulty ventilation did not usually lie with an insufficient volume of air. Other faults were much more frequent, and chief among them was artificial overheating.

When my studies were begun the almost universal plan of ventilating railway cars was to introduce fresh air by opening small windows at the top of the car which are usually referred to as deck-sashes. Sometimes large volumes would enter through these openings, sometimes little or none, and sometimes they would act as outlets only. With changes in the direction of the train or the wind these various actions would alternate. The openings being large, when they became active intakes, uncomfortable downward draughts were produced; when they ceased to act as intakes the total air supply would be much restricted and overheating would be readily brought about. The result of this plan was that ventilation was very irregular, that an adequate air supply could not be depended upon, and that comfortable or hygienic thermic limits could not be continuously maintained.

A little experimentation soon demonstrated that sufficiently large volumes of air for all the practical purposes of sleeping car ventilation will, under certain circumstances, enter the car through the several hundred feet of crevices about doors and windows. Thanks to the imperfection of the car builder's art, it has not been found possible—though much pains have been taken in trying—to build a railway car so tight but that much leakage may take place. When this occurs the air of the car remains relatively pure, heat does not accumulate, and comfort is maintained. It was therefore deemed advisable to utilize these crevices as air intakes and to attempt their regulation by providing a constant forced exhaust from the top. To this end a device was applied to the roof of the car which utilized the power supplied by train momentum to produce suction, thereby drawing air out through the deck-sashes formerly used as intakes.

The purpose aimed at was well achieved. The result was as contemplated. The average air supply was much increased and the

flow regulated. Perfect regularity of air supply was not realized; but it is one of the fallacies of general opinion concerning ventilation that perfect regularity of air supply is desirable. It has been pointed out by Leonard Hill that one's comfort depends in a vast degree on the stimulation of a changing physical environment—that it is due to the ceaseless variation of the temperature and motion of the surrounding air acting on the great field of cutaneous sensibility. The body is wearied by monotony; it is exhilarated by the train of reflex activities, beginning with vasomotor adjustments and ending with changes in tissue metabolism, which are set up by stimulating the skin with cold. In this light irregularities of air supply and air temperature become of great importance, but they should not vary beyond the limits within which comfort lies.

The objection is sometimes raised that simple exhaust ventilation, without the provision of inlets, cannot possibly furnish to railway cars a sufficient air supply. I, too, was originally of that opinion; but so far as the requirements of sleeping cars are concerned it was a mistaken opinion, as has been amply demonstrated. The quantity may at times be quite astonishing. I have measured up to 30,000 cubic feet per hour passing out through the ventilator duct from a small state-room with the door and windows closed, and when, therefore the air had to find its way in through crevices. Much of it came from the adjoining passageway, through the free-way under the door. Natural crevices are sufficient air inlets if properly utilized, and they possess a vast advantage over larger openings. The incoming jets of cold air are individually so small, and they take on such irregular motions in mixing with the stiller air within, that all the good effects of variegated temperature and motion are obtained without the disadvantages of uncomfortable cold draughts.

If a few large inlets are used and the weather is cold, the incoming air must be pre-heated. Pre-heating has its drawbacks, but it is necessary for most heavily occupied places. If sufficient crevices are available, and the occupancy is not too great, pre-heating may often be dispensed with. Few enclosures are so richly endowed with crevices as is a railway car; no other place is so constantly subjected to high winds, which cause crevices to let in outside air whether we will or not. In entering, these cold jets meet the warm stream flowing upward past the windows and the mixing begins at once, as it should. But whereas these crevices act irregularly, and in the main inefficiently, as air intakes when unaided, they act efficiently and with a sufficient degree of regularity when assisted by exhaust ventilators located at the top of the car.

Of course we cannot directly measure the amount of air flowing into a car through its many crevices. But we can measure the outflow through the ducts of exhaust ventilators, and we know the one must equal the other when only crevices are available as inlets. We can also estimate the air interchange by determining the proportion of carbon dioxid in the inside air and counting the number of occupants concerned in producing the contamination. Both these procedures have been carried out. By the former it has been demonstrated that the ventilators now almost universally installed on Pullman cars will each discharge from the running car some 12,000 or 15,000 cubic feet of air per hour. Six are applied to the twelve-section body of the standard car, and they all work constantly, though not at a constant rate.

But determining the outflow of air records only a minor fact. Without a study of the entrance, the distribution, and the interchange in the zone of occupancy it remains of little value. Herein lies the necessity of learning the carbon dioxid content of the air. It enables us to determine these things, and it must always be a part of any adequate study of air conditions. On it we may base also reasonably accurate estimates of total air supply; but it should be kept in mind that estimates so made will usually fall below rather than above the actual.

A fairly extensive study of carbon dioxid in the air of sleeping cars has been made, the details of which have all been reported in other places. We may refer here to a few general averages only which will be recorded in the form of charts.

In Chart I are represented the average air supplies, as determined from the CO_2 in the breathing zone, while cars were running in regular service at ordinary speeds and with various possibilities as to air inlets and outlets. The significant bars of the chart, representing cars where no intakes in addition to crevices were provided, are the first, fifth and seventh. Unaided by exhaust ventilators the average air supply through crevices alone was only 18,500 cubic feet per hour; aided by exhaust ventilators the averages were 40,600 and 53,000 cubic feet per hour for wooden and steel cars respectively. The difference between the latter two is believed to depend on the almost total absence in the steel car of crevices in its upper portion by reason of the absence of deck-sashes and a consequent absence of short-circuiting of air currents from deck-sash crevices to ventilators necessarily close at hand. There is in consequence a more constant withdrawal from below, and a more rapid changing of the air of the lower levels. The average occupancy of the cars represented in Chart 1 was about fifteen people

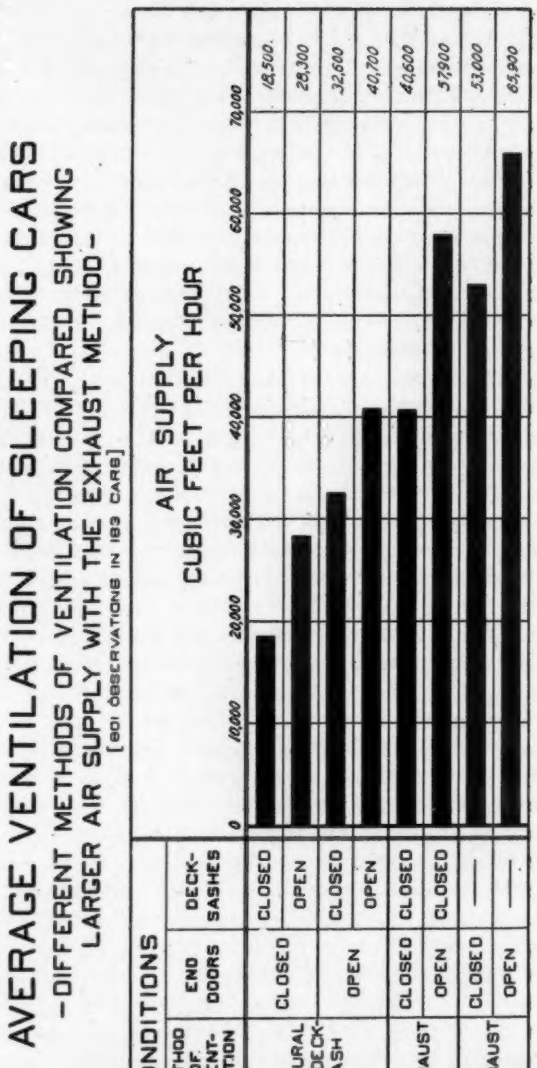

AVERAGE VENTILATION OF SLEEPING CARS
-DIFFERENT METHODS OF VENTILATION COMPARED SHOWING LARGER AIR SUPPLY WITH THE EXHAUST METHOD-
[801 OBSERVATIONS IN 183 CARS]
CONDITIONS
TYPE OF CAR
METHOD OF VENT-ILATION
END DOORS
DECK-SASHES
AIR SUPPLY
CUBIC FEET PER HOUR
0
10,000
20,000
30,000
40,000
50,000
60,000
70,000
WOOD
NATURAL OR DECK-SASH
CLOSED
CLOSED
18,500.
OPEN
28,300
OPEN
CLOSED
32,600
OPEN
40,700
EXHAUST
CLOSED
CLOSED
40,600
OPEN
CLOSED
57,900
STEEL
EXHAUST
CLOSED
53,000
OPEN
65,900

Chart 1

each, from which it follows that there was considerably more than 2,000 cubic feet of air per hour for each occupant in the cars equipped with exhaust ventilators, while there was only about 1,200 cubic feet per occupant in those not so equipped. Since the figures apply only to the twelve-section body, the practical maximum occupancy is about twenty-five people.

In Chart 2 are shown the relative average air supplies per person, as determined by the CO_2 content, for the upper berth, lower berth, and aisle of cars ventilated by open deck-sashes, and of wooden and steel cars ventilated by the exhaust method with only crevices as inlets. All berths were occupied by one person each, and each group of cars contained an average of about sixteen people. The air supply is seen to be considerably larger with the exhaust type of ventilation; and it is sufficient to meet the physical as well as the chemical demands placed upon it. Not only is the average volume of air supplied to berths increased by the exhaust method of ventilation, but the flow is more regular and is more constantly maintained.

It seems beyond question that a great improvement has been made in the ventilation of sleeping cars by the adoption of the simple exhaust method, and that the results are on the whole quite satisfactory. Carbon dioxid never rises so high—it is rarely as much as ten parts in 10,000—as to indicate an air supply insufficient for the needs of the body; the supply of air is reasonably uniform, while the innumerable eddies caused by minute incoming streams of cold air striking the warmer air within bring about thorough mixing and good distribution without destroying the ceaseless small variation so essential to continued comfort. The possibility of overheating still remains, and must always remain with any system of car ventilation as a matter requiring intelligent attention. But the assurance of a constant and fairly regular supply of cool air from without makes it a much less probable occurrence, while changes in car heating plans and simple instructions to car operators concerning the significance of thermometers have also aided greatly in remedying the evil.

III.

In the general application of the exhaust principle to sleeping car ventilation, a type of ventilator was adopted which, as had been proven experimentally, would operate constantly while the car was in motion and would discharge air at an average rate of about 15,000 cubic feet per hour at a 40-mile train speed, but with rather wide variations from time to time depending on the speed of the train

RELATIVE VENTILATION OF UPPER BERTH LOWER BERTH AND AISLE

-METHODS OF VENTILATION COMPARED-

(2746 OBSERVATIONS IN 130 CARS)

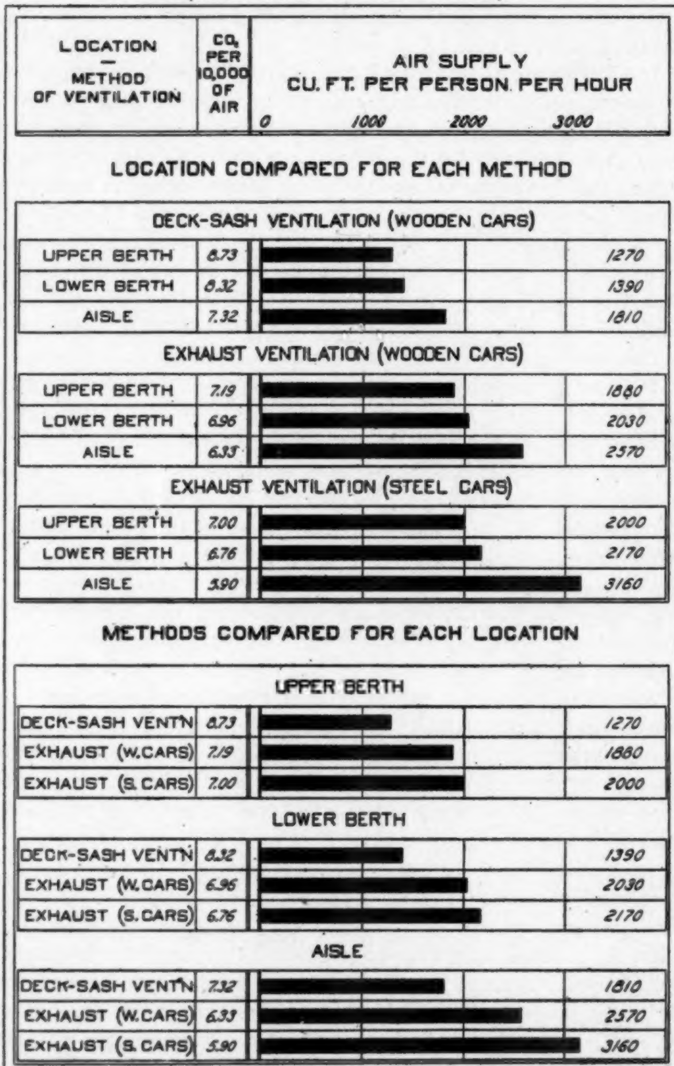


Chart 2

and the direction of the wind. This ventilator, which will be referred to as Type A, is now in use on the vast majority of Pullman cars. From time to time other types have been submitted by manufacturers or inventors, and have been applied to cars for testing against Type A, in order to determine which will work most efficiently in actual service.

The three important points in the comparison of exhaust car ventilators are: constancy of discharge of air, uniformity of action on the two sides of the car, and capacity. Observations concerning these points for all the ventilators so far tested were made while two cars, one equipped with Type A ventilators and one with the type to be compared to Type A, were operating adjacent in the same train. Two procedures were carried out: anemometer measurements of the flow of air through the ventilators, and determinations of the carbon dioxid in the air of the zone of occupancy.

Anemometer measurements were made successively and in series in each car, and included at least one ventilator on either side. The train speed was always recorded while a measurement was being made. It has been found that the capacity of exhaust ventilators, in any large group of averages, increases in direct ratio to the train speed, but that the two sides often act unequally on account of the influence of side winds. The measurements were always made with doors and windows closed, but with a drop-sash in either end door lowered 2¾ inches. Air samples for determining the CO_2 were collected with these drop-sashes also closed.

Carbon dioxid determinations were made in series from air samples collected simultaneously in the two cars at two-minute intervals. These samples were collected in rubber bulbs holding about 1000 c.c. while walking up and down the main aisle and were immediately transferred to bottles with waxed glass stoppers for transportation to the laboratory. The collection occupied about half a minute and was made at about four feet from the floor. Fair averages must therefore have been obtained.

The results of these comparative tests can be shown with sufficient accuracy and detail in a series of charts, thus avoiding the repetition of long lists of figures. For obvious reasons the various ventilators referred to are designated by letters. No attempt is made to describe them, since it was not my object to investigate the mechanics of exhaust ventilators, but was only to obtain evidence on which to base an opinion as to which one of the various types submitted was best adapted to our purpose of furnishing a constant air supply to sleeping cars and maintaining the changes so important to continued comfort and to a hygienic environment.

Ventilators were applied in the constant ratio of one to every two sections. With one exception (Type A-2) tests were made in standard steel cars.

In Chart 3 is represented the first test. Black dots in the upper figure show the individual observations in the car with Type A ventilators, open circles those in the car with Type B. Both are reduced from the one or two minute actual measurements to the equivalent rate per hour and plotted at the train speed observed while the measurement was being made. The diagonal lines show the general averages of air discharged per ventilator for each of the two types as marked at the right. Where a different number

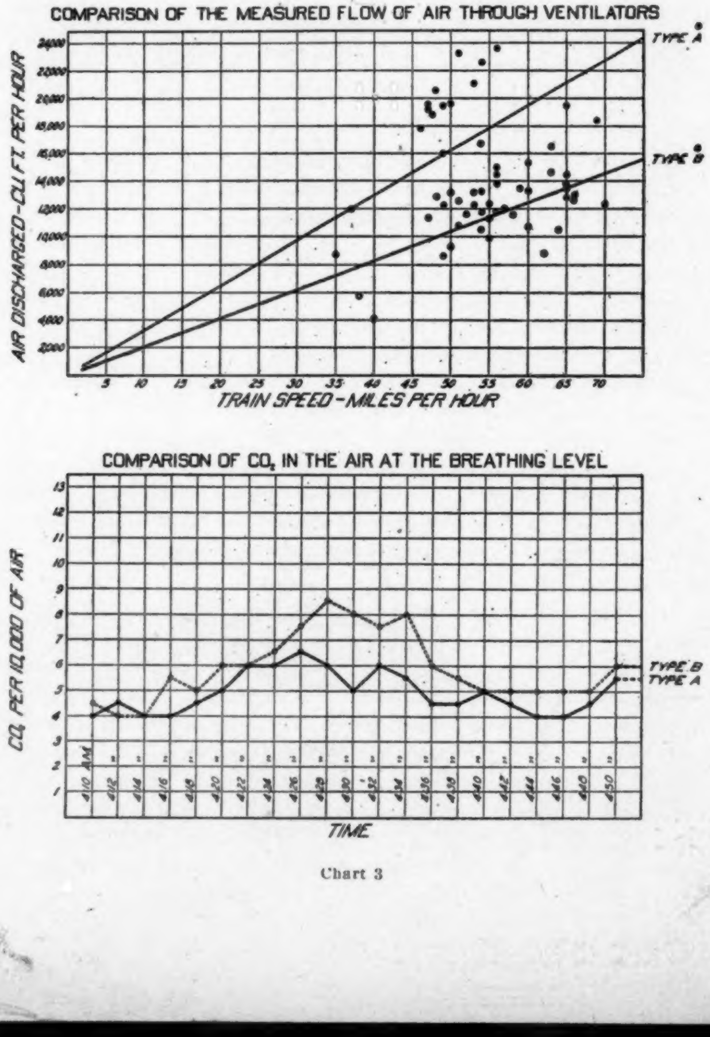


Chart 3

of tests was made on the two sides of the car, this general average is the mean of the averages for the two sides.

A glance at the chart will show that each type of ventilator acted with a considerable degree of irregularity as compared to its general average. This variation was somewhat greater for Type A, but Type B fell considerably below Type A in average capacity. The significant figures of the comparison are as follow, all ventilator capacities being reduced to the equivalent volume per hour at a 40-mile train speed so that they may be directly compared:

	Type A	*Type B*
Maximum	18,200	12,400
Minimum	7,000	4,100
Right side average	10,700	8,100
Left side average	15,900	8,700
Average of two sides	13,300	8,400
Relative capacity	100	64

In the lower figure of this chart the CO_2 determinations are shown. It will be seen that the carbon dioxid in the car equipped with Type B was almost constantly higher than it was in the car with Type A, as would be expected, provided the cars were equally occupied, from the greater discharge of air from the latter, but that the two curves run nearly parallel. The great rise in the middle of the curves was occasioned by very slow running of the train through a short interval. Such slowing, or a stop, reduces the interchange of air and always manifests itself in this way. A similar rise of CO_2 will be seen in other charts. The significant comparative figures follow:

	Type A	*Type B*
Average CO_2 per 10,000	4.93	5.88
Number of passengers	15	14
Equivalent ventilation	96,000	45,000

On the basis of this test Type A takes precedence over Type B on the score of capacity only.

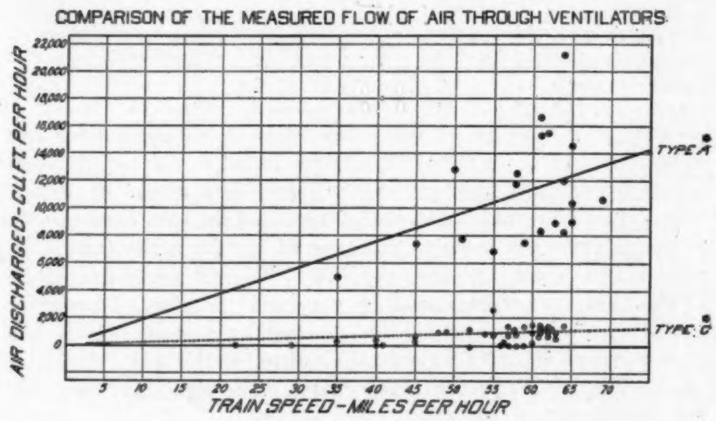
In Chart 4 a similar comparison between Type A and Type C ventilators is made. The significant figures follow, with the same reduction of ventilator capacities to the 40-mile train speed basis:

	Type A	*Type C*
Maximum	13,300	1,980
Minimum	5,000	0
Right side average	9,300	880
Left side average	6,000	490
Average of two sides	7,650	690
Relative capacity	100	9.1

Average CO_2 per 10,000	5.15	6.17
Number of passengers	14	14
Equivalent ventilation	73,000	38,000

Type C Ventilators were applied in pairs set close together, one in front of the other. The rear one of these was generally inactive as an exhaust and often allowed of backflow of air. The figures in the above table of capacities include only those ventilators actually working most of the time.

Type A in the above test had a capacity only about one-half of the usual finding. It is probable that adverse winds caused it to work at a low rate, though part of the apparent deficiency is believed



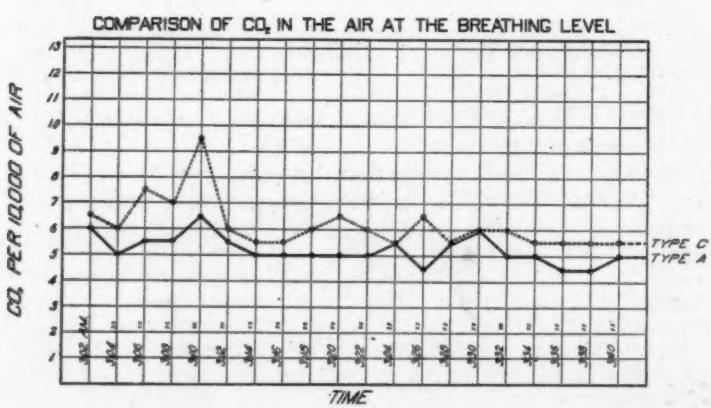


Chart 4

to be due to the method of measuring. The usual plan was to measure the flow of air within the panel opening to the ventilator duct; in this case it was measured inside of a screen covering this panel, consequently at a point of lower velocity than within the opening, though the area of the opening was utilized in computing volumes of discharge.

The CO_2 curves are consistent with the anemometer measurements, but lie closer together than would be expected. This is one of the instances where practically unaided crevices (Type C) furnished a fairly large air supply.

Chart 5 illustrates the test of Type D. Wide discrepancies are noticed between an upper and a lower group of observations belong-

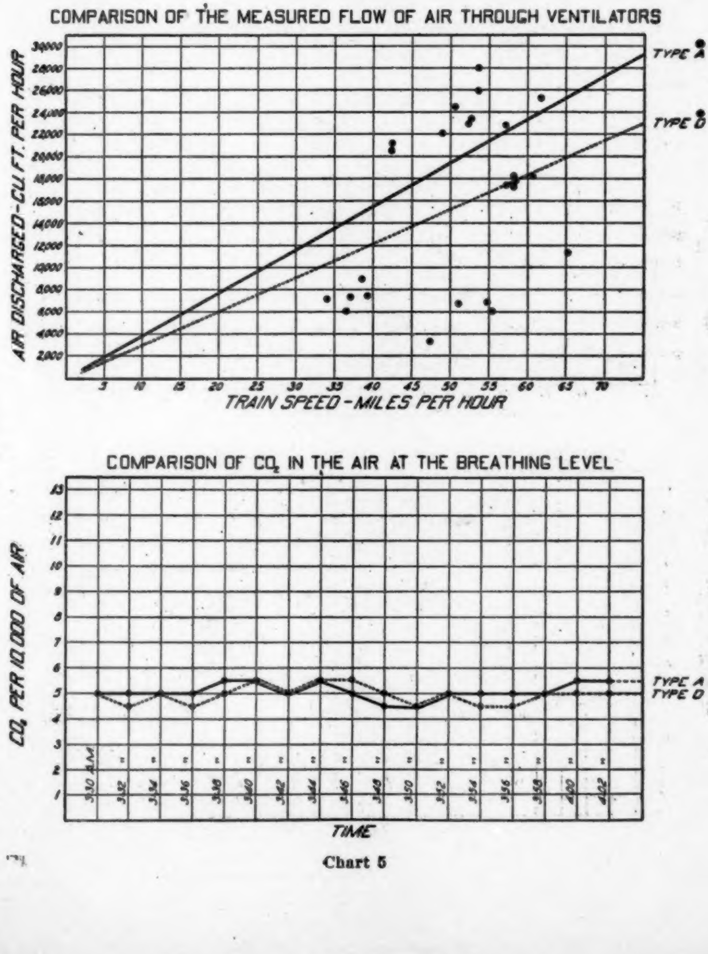
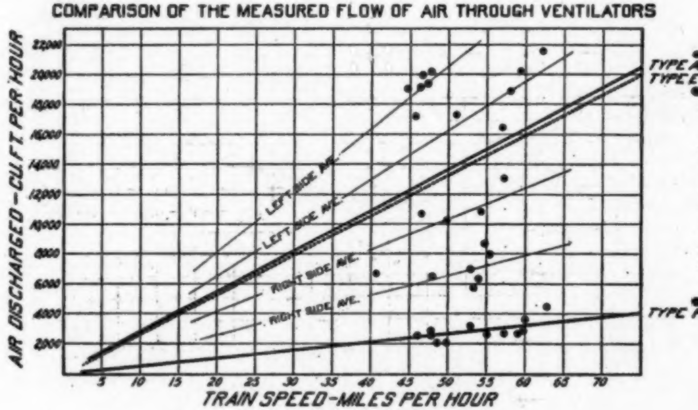


Chart 5

ing to this ventilator, the two groups representing right and left sides of the car respectively and indicating interference by side winds. Two similar groups will be noted for Type A, but they lie very much closer together. There was an occasional reversal of the air current in Type D located on the left side of the car. Comparative figures follow:

	Type A	*Type D*
Maximum	20,900	19,800
Minimum	12,000	2,800
Right side average	19,340	17,740
Left side average	12,120	6,240
Average two sides	15,730	12,210
Relative capacity	100	78.3

COMPARISON OF THE MEASURED FLOW OF AIR THROUGH VENTILATORS



COMPARISON OF CO_2 IN THE AIR AT THE BREATHING LEVEL

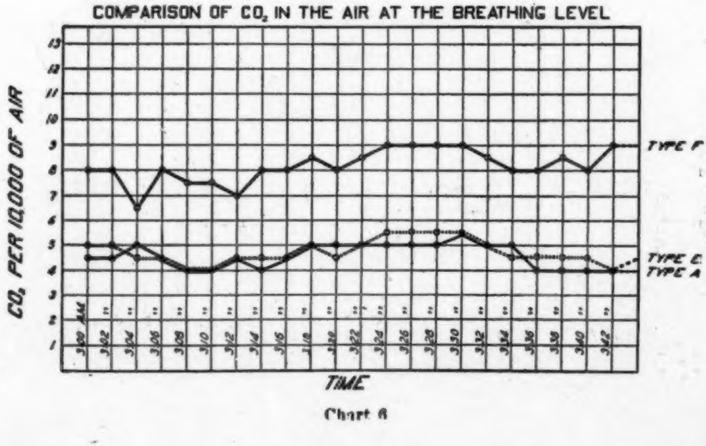


Chart 6

Average CO_2 per 10,000	5.15	4.94
Number of passengers	14	10
Equivalent ventilation	73,000	63,000

It will be seen from the chart and the averages recorded in the table that Type D is inferior on account of the great interference with its action produced by side winds. The relative ventilation of the two cars as computed from carbon dioxid is approximately the same as the relative capacities of the two ventilators.

Chart 6 records comparative tests of Types A, E and F, to which the following figures apply:

	Type A	*Type E*	*Type F*
Maximum	14,600	17,500	2,800
Minimum	6,500	4,300	1,570
Right side average	13,300	16,100	2,390
Left side average	8,500	5,200	1,790
Average of two sides	10,900	10,650	2,090
Relative capacity	100	97.7	19.8
Average CO_2 per 10,000	4.59	4.73	8.16
Number of passengers	7	10	10
Equivalent ventilation	71,000	82,000	15,000

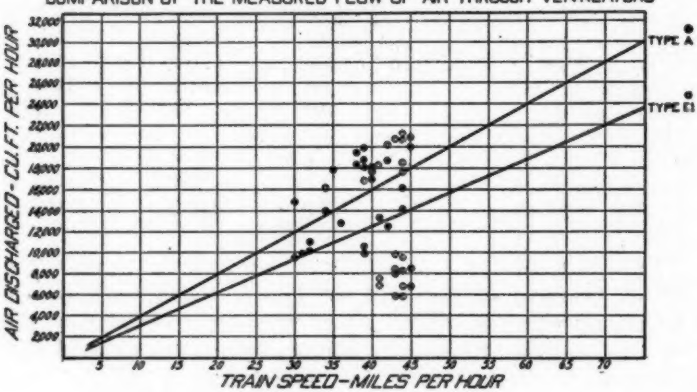
The capacity of Types A and E are essentially the same on the basis of measured air discharge but Type E shows much greater interference from side winds. As between Types A and E on the CO_2 basis, because of the light occupancy of the cars, no valid conclusion can be drawn further than to say that the occupied level of each was very effectively ventilated. For Type F it is clear that unaided crevices did not here prove adequate.

Chart 7 compares Types A and E-1, a modified form of Type E. Again we find a wide variation on the two sides, as shown by the two groups of circles representing the measurements of E-1 in the chart. The comparative figures are as follow:

	Type A	*Type E-1*
Maximum	21,100	19,200
Minimum	10,100	5,300
Right side average	19,100	18,200
Left side average	13,100	7,100
Average of two sides	16,100	12,500
Relative capacity	100	78
Average CO_2 per 10,000	7.54	7.54
Number of passengers	15	17
Equivalent ventilation	25,400	28,800

During most of the hour consumed in collecting air samples for this test the train was running very slow. This would result in lowering the interchange of air and in a higher average CO_2. Apparently the relation of ventilator capacity is not borne out by the computation of air supply made from CO_2 determinations. There were, however, six Type E-1 ventilators being used on a 12-section car, whereas only four of Type A were in use on a 10-section car. If we compare the ventilation as determined on the CO_2 basis with the theoretical air removal by four Type A and by six Type E-1 ventilators, the relation established on the basis of anæmometer measurements is still retained, E-1 showing about 80 per cent. of the efficiency of A.

COMPARISON OF THE MEASURED FLOW OF AIR THROUGH VENTILATORS



COMPARISON OF CO_2 IN THE AIR AT THE BREATHING LEVEL

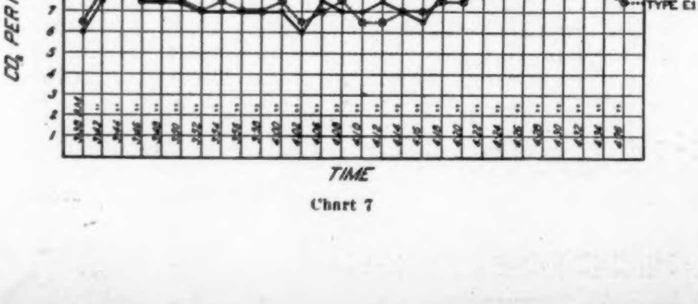
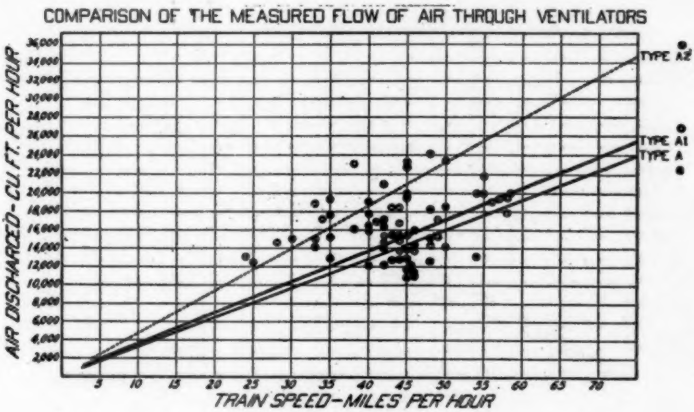


Chart 7

Chart 8 records the comparison of two modifications of Type A with Type A itself. Both Types A-1 and A-2 are seen to exceed Type A in capacity and to have at least as regular a flow. Type A-2 was on a wooden 10-section observation car of older type. The presence of loose-fitting deck-sashes may have given opportunity for much leakage at the top and so be responsible for a part of the increased outflow from this car. For this reason, and for other reasons not here indicated, the favorable results of this test for A-2 are not believed to represent a fair comparison, though the flow was maintained in this instance with great regularity and increased roughly in direct proportion to the increasing train speed. The comparative figures for the three ventilators follow:



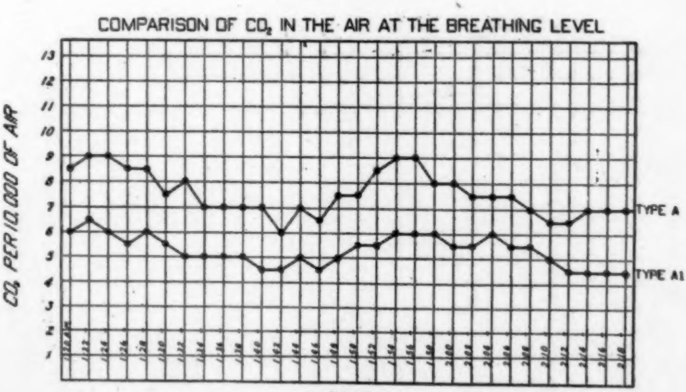
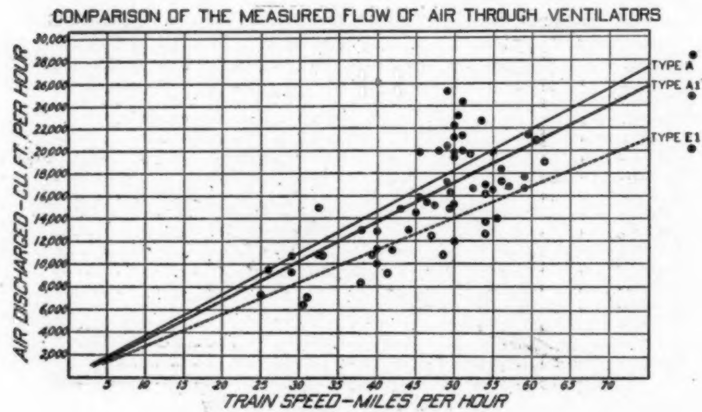


Chart 8

	Type A	*Type A-1*	*Type A-2*
Maximum	17,100	16,700	24,300
Minimum	9,600	11,400	14,800
Right side average	13,500	13,400	19,900
Left side average	11,800	14,000	17,600
Average of two sides	12,700	13,700	18,700
Relative capacity	100	108	147
Average CO_2 per 10,000	5.30	7.57	......
Number of passengers	7	19	.....
Equivalent ventilation	32,300	31,900	......

Carbon dioxid determinations were made for only two of these three cars, those having Types A and A-1 ventilators. In spite of



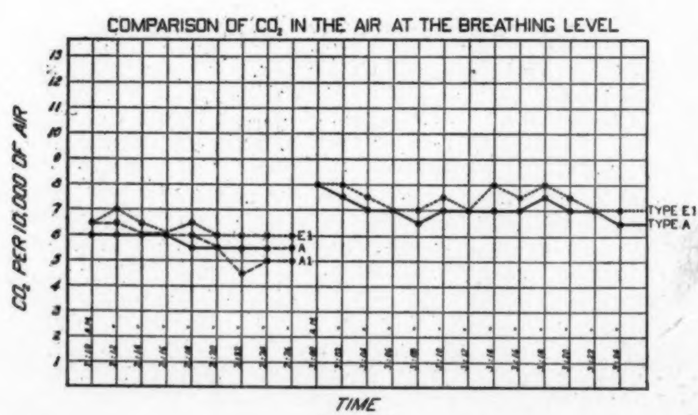


Chart 9

the fact that the curves lie far apart they are indicative of approximately equal air supplies for the two cars. The figures are, however, not dependable. After completing the analyses a leak was discovered in the apparatus being used which had the effect of making all CO_2 determinations too high. The leak was very slight, and we may suppose that all determinations were affected in approximately a like degree. The curves are retained in the charts because their relation to one another is significant and is probably correct, though they are not reliable guides for computing the effective ventilation. The same trouble is probably responsible for the high CO_2 of the last preceding test since the two were made near the same time.

The next and last comparison to be recorded concerns Types A, A-1, and E-1, and is presented in Chart 9. The figures follow:

	Type A	*Type A*-1	*Type E*-1
Maximum	20,700	18,500	15,200
Minimum	10,000	11,300	8,500
Right side average	11,900	14,800	10,000
Left side average	17,400	12,800	12,500
Average of two sides	14,700	13,800	11,300
Relative capacity	100	93.8	76.9

Average CO_2 per 10,000,			
1st series	5.72	5.65.	6.28
2nd series	7.12	....	7.46
Number of passengers	21	18	22
Equivalent ventilation,			
1st series	73,200	65,400	57,800
2nd series	40,200	......	38,100

Contrary to a previous observation (see Chart 8) it will be noted that the average delivery by Type A-1 in this test fell a little below Type A, but that, as previously seen, the various measurements of Type A-1 kept a little closer to the mean.

The air supplies as computed from carbon dioxid bear the same relation for the three cars as do the measured air deliveries. The higher CO_2 and the lower ventilation equivalent for the second series, in which only two of the cars were included, depend on slower running during this time.

In Chart 10 are presented the relative average capacities for air delivery developed in the preceding measurements of the various ventilators tested, each being compared to the capacity of Type A while operating in the same train, and therefore under as nearly equal conditions as are practically obtainable in service. It is

TYPES OF EXHAUST VENTILATORS COMPARED

—EACH TYPE TESTED AGAINST TYPE A, WHILE TWO CARS EQUIPPED WITH THE TWO TYPES OF VENTILATORS WERE OPERATING IN THE SAME TRAIN—

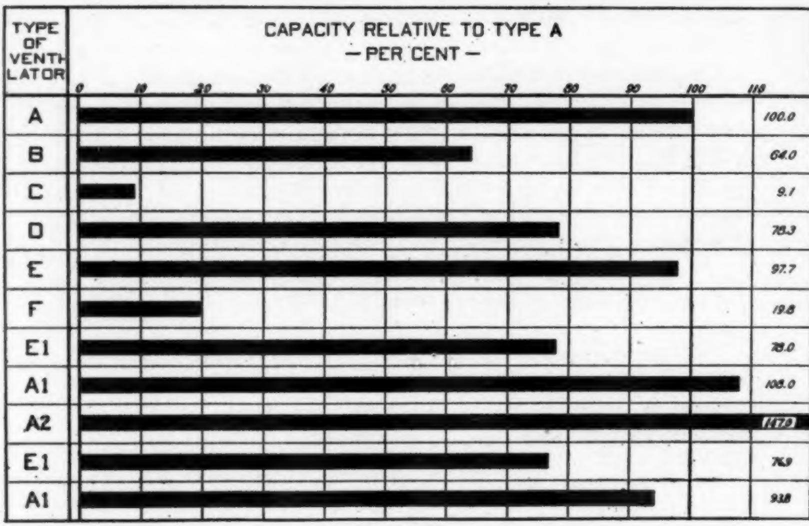


Chart 10

seen that Types A-1 and A-2 are the only ones that developed a higher efficiency than A. A second test of A-1 brought it a little below, as did also a third test not included in this series. For reasons already stated the comparison of A-2 is not considered reliable. While the delivery of Type E was essentially equal to that of Type A, the wide difference between the capacities on opposite sides of the car make it a less desirable device. The others all fell so far short in the amount of air delivered as to make the comparison unfavorable to them.

None of these devices will develop in actual service a capacity equal to that brought out in laboratory experiments, for the reason that the supply of air to them is restricted by its difficulty of entrance. The exhaust ventilator increases the entrance of air to a car rather indirectly than directly. The slight suction developed provides rather for the ready, constant, and immediate removal of air forced in by the strong winds to which a running car is constantly subjected than for enforcing its entrance. Regularity of action and a minimum of interference by unfavorable side winds is therefore an important point in their comparison. In this feature Type A or one of its modifications seems to be as good as any of the others,

and better than most, as can be readily seen by glancing over the charted measurements.

The foregoing seems to warrant the conclusion that no gain in the efficiency of sleeping car ventilation would be made by substituting for the present standard exhaust ventilator any one of those which have been subjected to comparative service tests. The volume of air discharged by the six of these applied to the main compartment of the standard car—amounting to some 50,000 to 100,000 cubic feet per hour—should be enough to maintain a sufficiently rapid interchange in the zone of occupancy for all the practical purposes of sleeping car ventilation. In my opinion it has been found so.

Synopsis of the paper

The paper explains at the outset that the popular conceptions of fresh air are erroneous.

It then describes the chemical properties and physical functions of air. It describes the recent researches and experiments relating to the function of respiration. It describes the method of testing ventilators on cars in actual service to get at the comparative value of the different types.

It illustrates by graphic charts the results of the tests.

DISCUSSION

Pres. Lewis: On behalf of the Society, I wish to extend our thanks to Dr. Crowder for this very interesting paper. We have not become acquainted with Dr. Crowder here as much as we have been in Chicago. I have known for a long time that the work he was doing was along lines of original research. I hope there will be closer relations between Dr. Crowder and the Society. The paper is now before you for discussion.

Mr. H. M. Hart: Just one question I would like to ask Dr. Crowder. That is, how he regulates the flow of air? For instance, some trains travel more rapidly than others and carry fewer passengers per car. I ask him if any attempt has been made to regulate the flow of air so that the draft is not excessive.

Dr. Crowder: No, no attempt has been made. What we attempted to do was to maintain a constant outflow of the air which is ordinarily forced in by the winds to which the car is subject when it is in motion. There is always a considerable in-flow, whether we desire it or not, and this can be maintained

and regulated if we provide for constant out-flow by exhaust ventilators.

Mr. Davis: I would like to ask if the direction of the wind does not have some effect on the amount of air going through the ventilators. For instance, if a car was going at forty miles per hour and had a fair wind with it at twenty miles, would not that make a greater difference, than if the wind was blowing against the train?

Mr. F. K. Davis: In regard to toxin poisons in the air. I remember a few years ago, Mr. Rosenwald made experiments in which he attempted to prove there were toxin poisons in the air. I presume the Doctor is following that line of work, and I would like to ask if anything further has been done along that line or if those experiments have been abandoned.

Dr. Crowder: I suppose you refer to the experiments of Dr. Rosenau. His results have not been confirmed by other workers. They have been repeated in New York by Weisman, and in London by Leonard Hill, and they have pronounced the positive results reported by Rosenau to be based on faulty technique. I think we may safely say that as things stand now, Dr. Rosenau's results are not to be considered conclusive. There is an interesting discussion to be brought up in this connection. It is not possible to take back into the blood the products contained in our expired air by breathing an atmosphere containing a slight excess of it.

If we rebreathe a little of the air we exhale, we breathe a little excess of carbon dioxide, and whenever we breathe an excess of carbon dioxide we breathe a little deeper; so we always take in just enough outside air to reduce the proportion of our hypothetical excretion in the air of the lungs to whatever may be its normal proportion for the individual concerned. The regulation is brought about by the CO_2, and the concentration of expired products in the air of the lungs is thus always kept at a normal uniform level.

Mr. McCann: I have spent a good many nights on sleeping cars, and I don't find so much trouble from the CO_2 content as the fact that I catch cold from the overheating of the cars, while sleeping. There should be something done to regulate the temperature of the cars. I think this is something that Dr. Crowder probably could assist us in materially. If he would advocate some kind of regulation to make excessive heat impossible.

A test was made some years ago and twenty-five roads were asked for their data for figuring the amount of radiation that goes into coaches and they varied all the way from 1½ sq. ft. per hundred cubic feet up to 8 sq. ft. and they were all running in the northern winter zone.

Dr. Crowder: Yes, the direction of the wind sometimes has a marked effect on the ventilation. One of the main points in the comparisons was to determine what type of ventilator is least affected by adverse winds. Of course, if we are running with the wind, we get less actual wind speed in relation to the motion of the train, and we get lower efficiency of the ventilation. If we are running with high side winds, then we find that the ventilators on one side of the car will work at a higher capacity than those on the opposite side. It happens, however, that this is largely equalized; where one side goes down the other goes up, so that our average capacity remains the same.

Mr. Davis: I think Dr. Crowder's results clearly show the advantage of exhaust ventilators for passenger trains. The results shown for Pullman cars that average about fifteen people, are very good, but the conditions in a Pullman car are the least of the evils we have to contend with. The ordinary passenger coach with perhaps an average of 80 persons in it is more of a problem, not only of air changing, but of heating. I find in traveling in the average passenger coach, there is a great tendency towards overheating. If the blast system of heating were installed we would be assured of fresh air and we would be assured of a more uniform temperature throughout the car and better control of the temperatures.

Mr. C. L. Riley: What happens to the sleeper that has to stand four hours on the side track?

Dr. Crowder: There is of course a much restricted air supply in that car. I have found that the carbon dioxide in the occupied berths of a car lying still for a long time will sometimes run up to twenty parts in ten thousand. This is about as high as it will go, and it usually stops short of this. I am sorry I have not some charts here which show exactly the findings in such cars. It is desirable that some additional means of providing fresh air to still cars be provided. Exhaust ventilators are the same as if there were no ventilators when the train is still, unless a very high wind is blowing that will cause them to operate. This does not usually occur.

We have put on some cars exhaust fans that work through the ventilating ducts in exactly the same way that the ventilator

itself works when the train is in motion, and then we find the carbon dioxide content of the air remaining at five or six parts in ten thousand instead of running up to fifteen. Instead of a rise when the train stops, the CO_2 then runs along practically on the level. It is also possible to keep the carbon dioxide fairly low in still cars by opening the end doors. If the proper temperature is maintained, however, there is no objection to the carbon dioxide running to fifteen parts in ten thousand.

Mr. Quay: It seems to me that the Pullman Co. ought to be able to furnish some better method of heating and ventilating the cars, they are far behind the times. In Cleveland where the street car company only charge three cents for riding all over town, most of their cars are heated by a heater in connection with a fan and they are very comfortably heated and ventilated. Does it not seem too bad that we cannot find some similar method for use in the Pullman cars with the prices they charge for their accommodation. Have often suffered from overheating in the Pullman cars and it is one of the most serious and uncomfortable things one experiences in traveling in them.

The method of heating the Pullman cars is very crude, uncomfortable and unhealthy. You are either exposed to cold drafts or have to breathe vitiated air.

Dr. Crowder: I would say in connection with this that there have been great changes in the method of heating Pullman cars in the last few years. They were formerly heated by the hot water system, but now they are heated by direct steam, and there are in each car at least seven units of control. With a little intelligent management of the valves, the cars can be heated to a comfortable point and not beyond. The real problem after all is only one of heat control and not one of air supply at all. Considering sleeping car ventilation in its broader aspects we need only inquire whether or not we are furnishing enough heat to be comfortable and not enough to be harmful. Sometimes we may furnish enough to be harmful, because it is always mildly harmful when it is uncomfortably hot.

I do not know of any better plan for heating cars than the one at present in use. If you gentlemen who have engineering proclivities can propose better ones, I have every reason to believe they will be carefully considered in all details as to their efficiency and desirability.

CCCLXXVI

VENTILATION OF INDUSTRIAL PLANTS

BY *C. T. GRAHAM-ROGERS, M.D., DIRECTOR, AND *WILLIAM T. DOYLE, MECHANICAL ENGINEER, DIVISION OF INDUSTRIAL HYGIENE, NEW YORK STATE DEPARTMENT OF LABOR

In 1907 among the labor laws was a section requiring proper and sufficient ventilation in factories. During the same year there was created the position of Medical Inspector of Factories and a campaign was begun by the Department of Labor of the State of New York for the enforcement of the section on ventilation. This action on the part of the Department undoubtedly did much to awaken interest in the question of atmospheric conditions, for previous to this action very little appeared in literature with reference to ventilation of industrial plants, while at the present day many agencies are at work, both in the scientific field and among the engineers to secure proper standards.

It is probably for the reason that, as the Department of Labor had so much to do with ventilation and most of the work fell upon the shoulders of the Medical Inspector, the Director of the division was asked to present this paper before the society. Many changes have occurred since 1907, new legislation has increased the scope of the section on ventilation, and special engineers have been added to the Department, but many of the problems still seem to be unsolved. Unlike the problem of the ventilation of dwellings, the ventilation of industrial establishments presents numerous difficulties, for we have both general and local ventilation problems to solve.

The Medical Inspector or, for that matter, any inspector having to do with industrial work, can do no more than ascertain the danger existing, and request the engineers to solve the problem, hence in presenting this paper, the major part has been contributed by the Mechanical Engineer of the Department.

We have conclusively proven the necessity for constant air changing in industrial establishments, and have noticed beneficial effects

upon the workers, also we have shown the necessity for efficient removal of dust, fumes, gases or vapors. The solving of these problems is not merely one of ordinary public health, but also of economics, for there is financial gain in many ways. The employer secures efficiency which enhances his financial returns, the worker secures better health with its financial returns, and the State secures healthier citizens which add to its prosperity.

In the list of papers presented at this convention for discussion, heating, lighting, and ventilating are treated from many specialized viewpoints, and much valuable knowledge and experience have been disseminated; yet in all the list there is not included a treatise on any one phase of industrial ventilation or lighting. Surely the reason is not because this field is not sufficiently extensive to warrant attention; 1,200,000 in this State alone are employed in the field of manufacturing on an average of nine hours each working day, representing a far higher number of man-hours occupancy of work rooms than any class of structures except tenements.

Neither may the omission be attributed to the ideal conditions generally prevalent in factory buildings with regard to the factors under discussion. A very limited examination of conditions will dissipate any idea of this being a fact. In no places other than manufacturing establishments will the ventilation problems present so many complexities and in no other class of structures will the solution thereof be found so primitive, if perchance any attempt at solution has been made. Aside from questions of temperature and air movement, the theatre, school or hospital furnishes no problems. These questions are of as great importance in manufacturing establishments, and in many of them there is added the problem of caring for injurious gases, fumes, vapors and dusts generated by the appliances and materials of manufacture.

In the matter of heating and ventilation of factory buildings the architectural profession is primarily at fault. In designing such structures the craft has been servient to the ideas of production superintendents and has depended on this latter class for guidance in matters of interior arrangement, professional ingenuity being exercised only in minor details aside from the exterior elevations. It has not consulted experts, as a class on the problems of ventilation, heating or lighting, because it has never understood the necessity of doing so, and the existence of the necessity has never been brought home in a way as impressive, to cite an instance, as the necessity of obtaining expert structural advice when the fashion in office buildings changed from the five-story to the twenty-five story type. No better evidence is required of the lack of co-operation be-

tween architects and ventilating specialists in the design of factory buildings than the absence of a set paper at this conference, relating to the problems peculiar to this class of structure.

The position occupied by the State Department of Labor relative to these pertinent questions concerning the welfare of the factory employee is and must continue to be supervisory. The purposes of the organization as prescribed by law is not to design and superintend the erection of factories, loft buildings, etc., or to pass on the adequacy of proposed plans for converting an ancient dwelling into one of these. Its province does comprehend the efficient ventilation and lighting of factories among many other details, and a conscientious effort is at all times made to administer the law in all respects. The eyes of the department are those of its inspectors, who though well qualified for this general work, are not experts in heating, lighting, ventilation or industrial economy. The orders issued by these inspectors to provide adequate light or ventilation are numerous. The appeals from these orders are likewise numerous, and not infrequently a case is carried into court against an owner who neglects or refuses to comply with orders issued against his plant. Sometimes the presiding judge sets himself up as an expert and rules that the light and ventilation afforded are adequate. Sometimes the owner makes the necessary changes or alterations under protest, and then thanks the inspector on a subsequent visit for having compelled him to do so. There are all sorts of combinations of these general instances. The facilities possessed by the department for educating the factory owner in matters such as these are very limited and may be likened to the imposing of a fine on a misdemeanant, the fine being the difference between the sum which the factory owner pays for the improvement ordered by the department and the amount which it would have cost him to make the same installation at the time of building if it had been included in the original plans.

It is seldom that a complete set of plans covering the proposed activities to be carried on in the different parts of a factory reaches the department. However, one such set was received not long since, and in checking the exhaust system shown on same, attention was drawn to the fact that there were soldering rooms, lacquering rooms and even buffing rooms in which no provision was made for caring for the fumes or dust generated. The attention of the designing engineer was called to these apparent omissions, and he explained that he had proposed a very comprehensive system of mechanical ventilation throughout the plant, only to have it bluepencilled indiscriminately by the owner. The indiscriminate pencil-

ling was all too apparent, as refinements had been allowed to remain, whereas essentials were excluded. There is a surprise coming to that owner on the occasion of the first visit to the plant of a department inspector.

It is necessary that an educational campaign be planned and carried to the notice of factory owners. They are generally of the type described by one author as not seeing beyond the penny of today; but who when convinced that better working conditions mean more money to his credit, is your convert. In planning his factory, he has left all the details which he considers unimportant to his architect, depending on the architect to provide the facilities of heat, light and ventilation, which he appreciates vaguely, but whose importance he does not understand. It is a duty of great importance to show the architectural brethren, and through them the owners, that expert advice in these matters will repay them many times over, in better workmanship, increased output, decreased time lost per employee through illness, decreased cost of illumination and in numerous other ways. It is a duty to the country to see to it that the next generation and the next after that are not sprung from starvelings, for while we, as a nation, have heretofore had the brawn of Europe from which to draw our manufacturing labor, we may well look to the day in our own land when the conditions so aptly described by Professor Arnold may obtain.

Synopsis of the paper

A dissertation on the factory inspection laws of the state of New York, explaining the difficulties of enforcing the recommendations of the inspectors.

Deals with the need of highly trained and specialized inspectors. Tells of the difficulties in getting adequate ventilation owing to original defects in the design of factories and shows that it is of financial advantage to a factory owner to provide adequate ventilation. Explains that the Department of Labor relative to the question of the welfare of the employee is, and must continue to be, advisory.

Speaks of the great need of an educational campaign to influence factory owners.

DISCUSSION

Mr. Feldman: An appeal to the architect as suggested by the writer of the paper is futile, because they would not pay any attention to it, but if there were laws passed compelling them to provide certain requirements as there are in some cities, as for

instance in the city of Cleveland, inadequate plans would not pass inspection. I would suggest that the Secretary look up records of all papers presented to the Society on Ventilation for Factory purposes, and give them more publicity.

Mr. Hart: I have made some rough computations regarding the cost in an industrial plant where fifty men are employed, I should say an air conditioning plant could be installed for $3,500 to take care of that many persons. If it increased the efficiency of the employees one per cent., it would be a good return on the investment.

In reference to the statement that "factory owners have been compelled to put in air conditioning apparatus and thanked the inspectors for making them do so" this is the experience that the Chicago Health Department are having right along, especially in the case of theatre ventilation, which is perhaps not so closely allied to the subject. Quite a large Chicago theatre put up a fight in opposition to the department. This department threatened to close them up if they did not put in a proper ventilating equipment. They did put it in and I did the engineering for it. About six or eight months later, I was called on the 'phone and the manager said he wanted me to visit the job, he said they were having a lot of trouble with the plant due to noise and he could not see how he could continue to run the plant with the excessive noise.

I went out and discovered that he had changed the pulley from the motor and increased the speed of the fan 50 per cent. He said that he was so pleased with the system that he thought he would improve it still more and was willing to burn the extra coal and use the extra current.

Secretary Blackmore: I want to say a word on the question of education. This Society has a great work before it, particularly in the line of publicity. Publicity cannot be carried on without money and the Society cannot get sufficient money unless it can increase its membership.

If each member would obtain one or two new members to add to the Society we would have an income large enough to carry on a vigorous campaign of education.

Mr. Baldwin: The City of New York is the greatest sinner in the question of "no-ventilation" that we have. The new Municipal Building, housing many thousands of city officers, and employees, is absolutely without ventilation as far as the offices, the anterooms and the private hallways or corridors connecting the suites of rooms are concerned. The first work in reaching the

general public should be to reach the officials of the city of New York on the question because we can make the city officials listen to us. I think this Society should take the city government in hand at once and force recognition on this subject.

Mr. F. K. Davis: Mr. President, speaking along the lines of the paper just read, and relating to what Mr. Blackmore has just said, we had with us last evening for a few hours, one of the leading architects of this country. He dropped in to spend the evening and said he would try to be back to-morrow. He wanted a couple of membership application blanks for the purpose of inducing two of his engineers to join the Society. We are making some progress.

CCCLXXVII

TESTS ON THREADING STEEL AND WROUGHT IRON PIPE

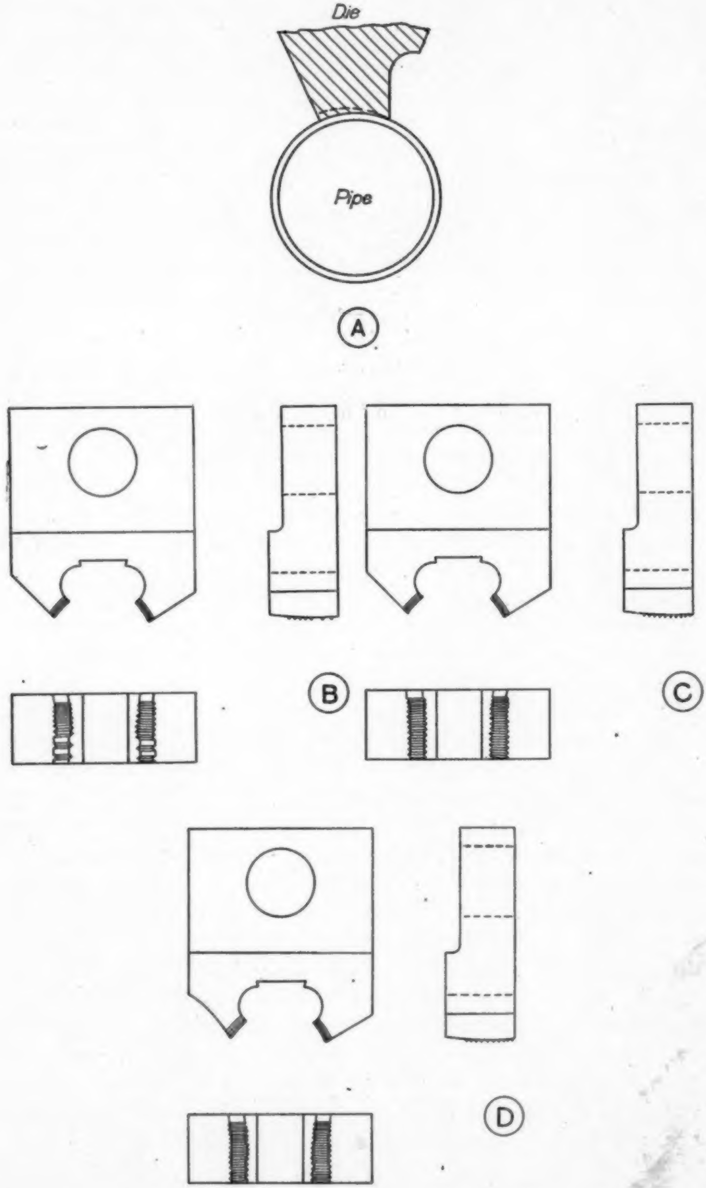
BY C. G. DUNNELLS*

Steel pipe was first manufactured in this country at Wheeling, West Virginia, about 1887. Previous to that time wrought iron was the material chiefly used in the manufacture of the smaller sizes of pipe used for gas, steam and water lines.

As in the instance of structural shapes, the introduction of steel caused a rapid decline in the production of wrought iron until, during the year 1913, 2,189,000 tons of steel pipe were manufactured in this country, as against 313,000 tons of wrought iron pipe. The steel pipe formed 87.5 per cent. and the wrought iron 12.5 per cent. of the total. There are two important factors that have caused this great growth of the steel pipe industry,—the lower cost of production and the improved physical quality of steel as compared with wrought iron.

By far the most prominent objection that has been raised against steel pipe by plumbers, steam fitters and other users has been the difficulty of threading when using hand stocks. At the Carnegie Institute of Technology the matter was brought to our attention by the fact that the students encountered trouble when cutting and threading steel pipe in our plumbing shop. In some cases the threads stripped off in spots, in other cases the pipe split in the weld. These difficulties led us to inquire whether the fault was in the pipe itself or in the dies which we had been using. With this idea in view, we undertook an extensive investigation of the factors involved in the threading of steel and wrought iron pipe, using various types of dies. Lengths of standard steel and wrought iron pipe, in sizes from 3/8 to 2 inches in diameter, were obtained from certain of the best known manufacturers, and stocks and dies were procured from jobbers handling pipe fitters' supplies.

* Head of Department of Building Construction, Carnegie Institute of Technology.


Die
Pipe
A
B
C
D

Throughout the investigation actual working conditions were duplicated as closely as possible, except that the pipe was held in the grips of a pipe-threading machine and revolved while the die remained stationary. The speed was 11 R.P.M., the length of the stock arm from center of pipe to scale hook was 2 feet 2 inches, the pipe projected 11 inches beyond the grips, and burrs were left after cuts were made. The force required to hold the die stock stationary was measured with a spring balance, (see illustration No. 1), the readings were all made by the same man and at the same times during the cutting of the threads, so that the results obtained are strictly comparable. Readings of the force required to thread are the average of from ten to fifty cuts for each test. The fact that more than 1,200 threads were cut will give an idea of the scope of the investigation.

The types of threading chasers used were classified as "A"—properly shaped receding chaser; "B"—Armstrong type skip tooth die; and "C"—Regular Armstrong type. By type "A" is meant a die with an inclination of the cutting edge to the surface of the pipe of at least 12 degrees and also possessing sufficient clearance to make it a good lathe tool. By type "B" is meant one which has a slight rake, some clearance, and which has alternate teeth omitted. By type "C" is meant one which has no rake and little or no clearance.

Comparing the force required to thread a 1½-inch steel pipe, type "B" required 58 per cent. and type "C" 80 per cent. more force than the lipped chaser type "A."

Suppose a plumber installing basin waste and vents in a modern office building cuts twelve pieces or twenty-four threads on 1½-inch pipe in a day with the old-fashioned solid die commonly used. By referring to Table No. 1 it is seen that he is expending 24 times 4,530 foot-pounds or 108,720 foot-pounds of unnecessary work, this being the excess of work required using the type "C."

For vacuum sweeper work fifteen pieces of 2-inch pipe are ordinarily cut and installed in a day by a plumber and helper. Tests No. 1 and No. 3 indicate 4,940 foot-pounds of unnecessary labor per cut, or 148,200 foot-pounds for the thirty threads cut in a day's work. At this time, when the cost of labor is far more than that of materials such as pipe and fittings, the necessity of efficient labor and hence efficient tools is strikingly evident. It is to be regretted that the trade has been slow to adopt improved dies which do a better quality of work and make as many cuts as the old-fashioned chasers with less wear on the dies, and with less force required to thread. During the progress of the tests we observed that the

Mild Steel Pipe

Test Number	Pipe Size (Inside)	Number of Cuts *	Type of Die	Force required to thread (Pounds). Minimum	Maximum	Average	Foot Pounds Work per Thread (Average)	Number of Threads per Cut	Total Foot Pounds of Work per cut	Percent Increase of Work over "A" Type
1	2	50	A	48	72	64	870	12.5	10875	
2	2	50	B	48	90	76.5	1040	12.5	13000	19.5
3	2	50	C	56	120	93	1265	12.5	15815	45.4
4	1½	50	A	32	42	36	490	11.5	5635	
5	1½	50	B	42	72	57	775	11.5	8910	58.1
6	1½	50	C	32	90	65	884	11.5	10165	80.4
7	1¼	50	A	26	48	39	531	11.5	6105	
8	1¼	50	B	34	64	52	707	11.5	8130	33.1
9	1¼	50	C	28	76	60	816	11.5	9385	53.7
10	1	50	A	20	24	21	286	10.75	3075	
11	1	50	B	24	36	30	408	10.75	4385	42.7
12	1	50	C	24	60	46	626	10.75	6725	119.0
13	¾	50	A	14	26	20	272	10.75	2855	
14	¾	50	B	15	34	29	394	10.5	4135	44.9
15	¾	50	C	16	40	33	449	10.5	4710	65.1
16	½	50	A	10	16	13	177	10.5	1860	
17	½	50	B	10	26	19	258	10.5	2710	45.7
18	½	50	C	12	32	23	313	10.5	3285	76.9
19	⅜	50	A	10	16	13	177	10.25	1815	
20	⅜	50	B	8	18	14	190	10.25	1950	7.3
21	⅜	50	C	10	22	18	245	10.25	2510	31.3
Solid Die after Lipping										
0	1½	10	D	26	79	51	694	11.5	7980	
Wrought Iron Pipe										
31	1½	25	A	24	44	40	544	11.5	6255	
32	1½	25	B	40	60	52	707	11.5	8130	30.0
33	1½	25	C	32	82	58	789	11.5	9075	45.0
34	1¼	25	A	26	32	30	408	11.5	4690	
35	1¼	25	B	28	54	44	598	11.5	6880	46.5
36	1¼	25	C	24	62	50	680	11.5	7820	66.7
Hard Steel Pipe										
30	1½	10	C	42	120	79	1074	11.5	12350	

Table No. 1

* **NOTE:** The term "cut" means one end of pipe.

maximum force was required to cut the first few threads with type "A," or lipped die. When the solid die was used, the maximum force was required near the end of the cut. This is a further advantage for type "A," because the workman's energy is greatest when he begins a cut.

By "hobbing" the solid chaser so that it possesses some lip and clearance, it was found that the simple operation reduced the required force 22 per cent. and also improved the quality of the threads.

Illustrations No. 2 to No. 4, inclusive, in which group numbers correspond with test numbers in Table No. 1, show results of threading tests. Groups 1 to 21, inclusive, are mild steel pipe, 31 to 36, inclusive, are wrought iron pipe, and No. 30 hard steel pipe.

Comparing iron and steel pipe we found that the force required to thread is somewhat to the advantage of the iron, as shown in the following table:

Type of Die	Force required to thread 1¼-inch pipe		Force required to thread 1½-inch pipe	
	Wrought iron	Steel	Wrought iron	Steel
A...............	30 lbs.	39 lbs.	40 lbs.	36 lbs.
B...............	44 lbs.	52 lbs.	52 lbs.	57 lbs.
C...............	50 lbs.	60 lbs.	58 lbs.	65 lbs.

This advantage is accounted for by the fact that the steel is stronger and tougher. It also gives a cleaner-cut, stronger thread because of the absence of oxides and cinder in pipe steel. A marked advantage for steel is had using the lipped die, there being many less slivered threads than in the case of wrought iron.

Using the hard steel pipe, the defect in which led us to make the investigation, we made several tests, the results of which are given in Table 1. These results show that considerable more force is required to thread this pipe than is required to thread either wrought iron or soft steel. Often it is a temptation to use the hard steel because of its cheapness, but experience has proven that it will not stand corrosion nearly as well as either wrought iron or soft steel pipe. The loss in labor by the extra force required to thread it and the large number of useless cuts will more than offset the difference in cost between it and soft steel.

At the present rate of wage, every bad cut means a considerable loss to the contractor. The ideal material is one which gives a maximum percentage of good cuts with a minimum amount of work. Hard steel for this reason is not desirable and can not be economically used.

One of the disadvantages of all welded pipe is the fact that it sometimes splits while being threaded or bent. To compare mild steel and wrought iron in this respect we conducted a series of tests, the results of which are given in Table No. 2. In these tests the pipe was held in the grips of the pipe-threading machine as in the threading tests. The projecting end of the pipe was securely held by dies whose teeth had been ground off. The stock was held stationary by a spring balance as before, and readings taken as the pipe was twisted. The maximum readings before failure are given in Table No. 2.

Mild Steel Pipe

Test Number	Pipe Size	Number of Cuts	Maximum Pull on 26" Lever Arm	Broke at Weld	Foot Pounds of Work per Twist
22	1½	10	510	4	3471·1
23	1¼	10	325	4	2212·2
24	1	10	255	3	1735·7
25	¾	10	160	4	1089·1
26	½	10	90	6	612·6
27	⅜	10	52	3	353·9
		Wrought Iron			
28	1½	10	320	4	2178·2
29	1¼	10	260	6	1769·8

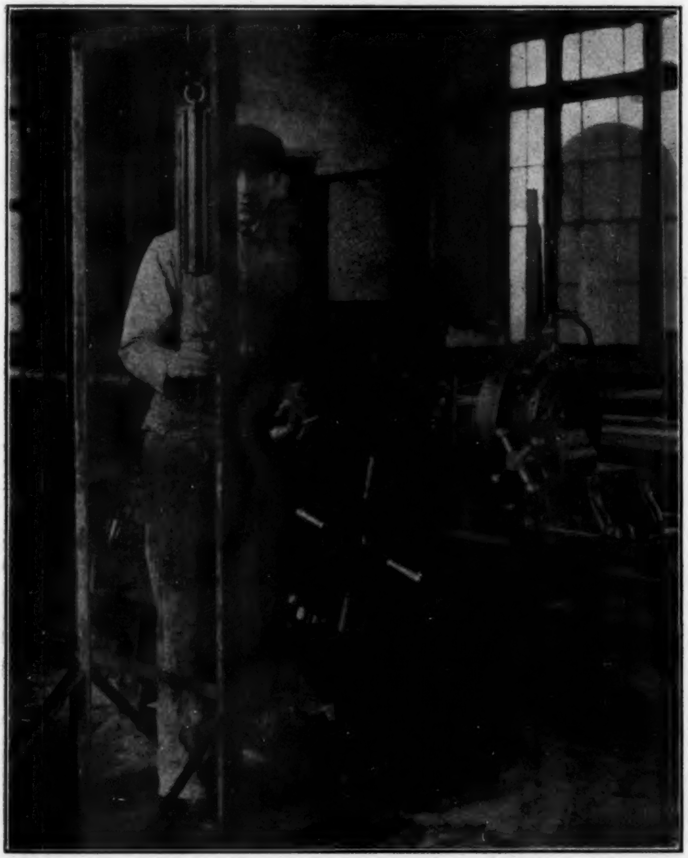
Table No. 2

Comparing these results we found that it took at least 25 per cent. more force to twist open the seam in the steel pipe than in the wrought iron. The force required to twist steel pipe to the point of failure was found to be so much greater than the force required to thread it that it is certain that there is little likelihood of soft steel pipe s failing under torsional stress while being threaded. The number of pipes opening in the weld in these twisting tests was about the same for soft steel and wrought iron, but the ultimate strength of the steel pipe was much greater than that of the wrought iron pipe.

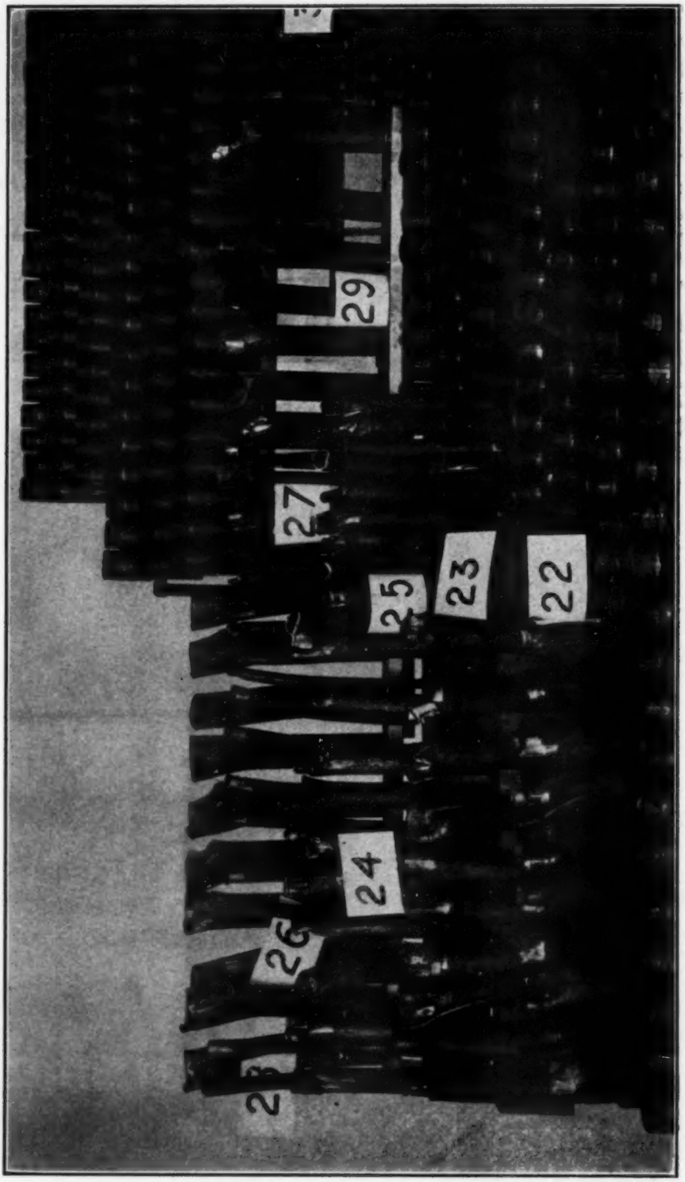
In making a careful study of the results of these tests, it is evident that the force required to thread soft steel pipe using type "A" die (the narrow, inserted, receding chaser) is much less than

the force required to thread wrought iron pipe with type "C" die which is commonly used. Several makers are now manufacturing the improved type of die and a considerable number are already in use.

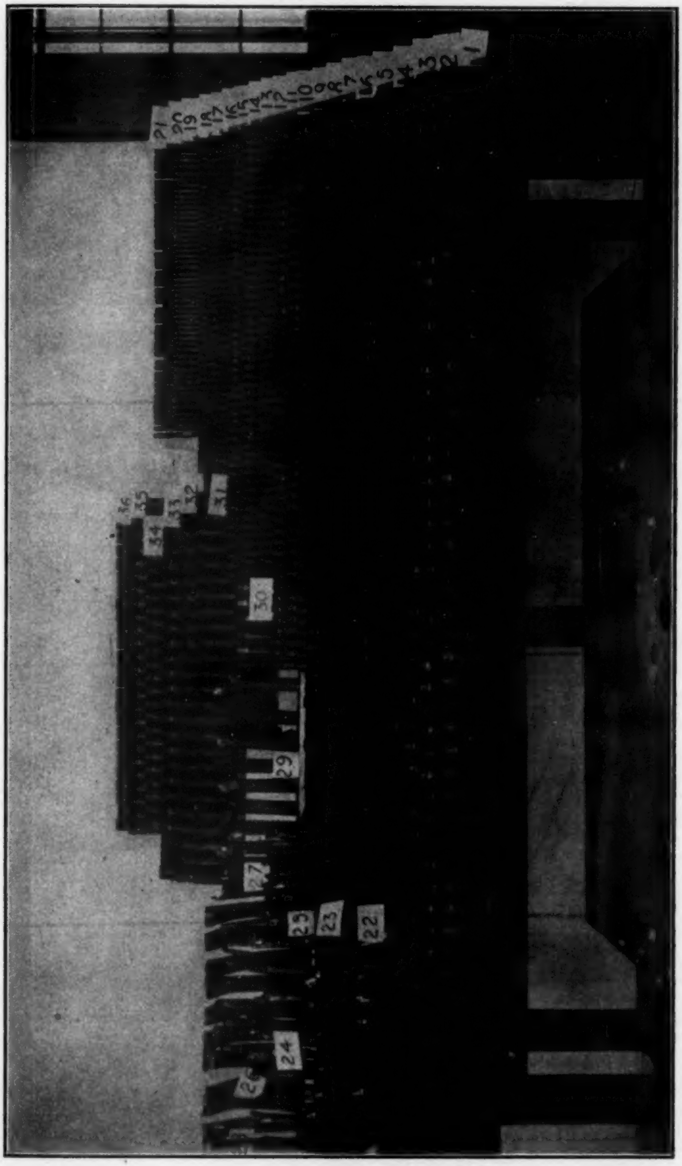
It is not the intention of this article to advocate any particular make of die, but to call attention to the best type for threading under present conditions. A glance at the results shows the marked



No. 1



No. 2



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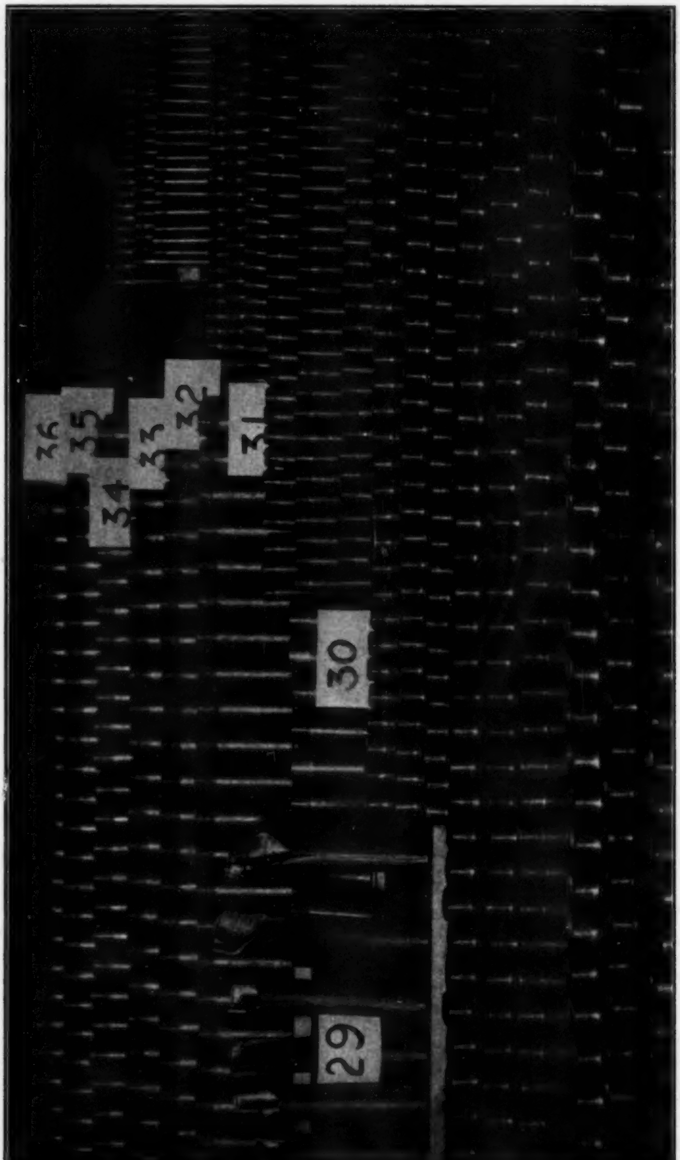


No. 4

saving in labor that can be effected by using the proper type of die. The practical man will appreciate the significance of these facts, for they greatly affect the quality and quantity of his work. Old-fashioned tools served very well in the past, but we have today reached a point where both labor and material are important. The cost of material fluctuates but little, while the cost of labor has a wider range. The success of the contractor, as spelt in dollars and cents, depends greatly upon his equipping his employees so as to obtain maximum efficiency.

Diverging somewhat from the subject of threading, it may not be out of place to mention the much discussed subject of corrosion. Several comparisons have come to our attention which may be interesting.

In July 1911, a gas line was laid at Camp Louise, Carnegie, a farm of several hundred acres belonging to the Institute and located about 30 miles from the city of Pittsburgh. At a convenient place in this line five lengths of 2½ inch black wrought iron pipe, the product of a well known and reputable iron pipe manufacturer, were alternated with five lengths of 2½ inch black steel pipe obtained from an equally well known steel pipe manufacturer. The location of the test lengths was so arranged that they would lie on the surface of a very marshy spot below a driveway. When this line was inspected last November, it was impossible to distinguish between the two kinds of material by the nature and length of corrossion. There was no appreciable difference between the wrought iron and steel, both showing a great number of small shallow pits uniformly distributed over the entire surface of the pipes. The absolute uniformity in the rate of corrosion of the two materials led us to the conclusion that in this respect mild steel is fully equal to the best wrought iron obtainable.

The rapid deterioration of a tin-lined brass hot water supply line in the Margaret Morrison Carnegie School brought to our attention the more corrosive nature of our present water supply as compared with water supplied ten to fifteen years ago. This pipe, furnished by one of the best known manufacturers of brass tubing, was installed about eight years ago, and is already giving our engineers cause for anxiety.

The fact that the use of steel pipe was rapidly increasing at the time when water supplies were gradually but certainly becoming more and more corrosive, has often led to the unjust conclusion that steel pipe, and not the character of the water, was responsible for the rapid deterioration of many recent pipe installations.

It is the duty of the engineer to specify the best material he can obtain when the costs of various materials are approximately the same. Our tests and observations have shown that mild steel is the most desirable pipe material now obtainable, the cost being considered. However, it is necessary, when using this material, to use tools adapted to it, and this done, many of the troubles connected with its use will vanish.

DISCUSSION

Mr. F. N. Speller: The investigation which led up to the presentation of this paper was carried on at the Carnegie Institute of Technology under the direction of Mr. C. G. Dunnells, head of Department of Building Construction, with the co-operation of Mr. Wm. A. Teemer, head instructor in plumbing, Department of General Equipment and Installation. They had noticed the usual difficulty in threading certain kinds of pipe with standard dies and determined to make a practical investigation. The materials used were purchased on the market and standard dies were used which anyone can obtain from jobbers' stocks. A little further data as to the three types of dies used might be useful:

Type "A" represents the class of die using the receding chasers, such as the "Toledo," "Beaver," and others. The chasers containing only about four teeth are relieved and usually provided with some rake to thread steel pipe, although this rake is not always sufficient. Very few, if any, of the die makers seem to follow up the use of their tools. Any investigations which they make as to the proper shape are apparently made on new tools and not as they should be on tools which have cut several hundred threads. The shape of the chaser has very much to do with the rapidity with which it wears. The receding type of chaser, which is coming so generally into use, has many advantages and makes for ease of cutting and gives much less friction.

In type "B" (this skip-tooth style) every alternate tooth is cut away to reduce the friction. This die is provided with relief and rake and usually cuts a good thread.

Type "C" is the ordinary "Armstrong" or "solid" type with which you are all familiar. The shortcomings of this die are quite apparent. It usually has no relief whatever and very little rake and insufficient chip clearance. All these points make for hard threading and a rough job.

In some experiments which we have been making on the tightness of standard threads we found very few threads cut by standard dies which could stand 100 lbs. steam pressure without leaking unless provided with some lubricant or "dope" to fill up the threads.

I would like to call attention to the material used by Mr. Dunnells. The iron and steel pipe were made by the largest Pittsburgh manufacturers. In addition to this, he had what he called "hard" steel pipe made by some outside concern. By the term "hard" the author probably means hard to thread. As a matter of fact, this pipe was probably softer than the other as it is the soft and tough steel that offers the most difficulty in threading. The larger manufacturers should be able to control this feature nowadays and as a rule are only too glad to hear of any difficulties in the threading question and fix the matter up by showing the consumer how to put his dies in proper shape.

Another advantage of easy cutting dies which might not be apparent at first is avoiding undue torsional strain on the pipe. By passing the elastic limit of the material in torsion the weld may be bent or strained so that it will fail in service under pressure. This may not be discovered until the pipe is installed and necessitates expensive repairs.

It is satisfying to note that since this question of threading dies was called to public attention by me several years ago there has been considerable improvement on the part of the die makers. At the same time there are a number of die makers who still pay little attention to the points referred to in this paper, and as these are factors that make for efficiency, a paper of this kind should be of the greatest practical value to those interested in the use of pipe.

At this point a series of interesting lantern slides were shown illustrating the various types of dies and the relative consumption of steel and wrought iron pipe from 1884 to the present time.

CCCLXXVIII

SOME PHASES OF ROOM HEATING BY MEANS OF GAS BURNING APPLIANCES

BY GEO. S. BARROWS*

The application of gas burning appliances for the purpose of heating rooms appears to be, at first glance, of the simplest character, limited solely by the operating cost in comparison with that of competing fuels. This very simplicity, which makes the use of such appliances almost universal may permit, however, abuses more or less serious and the object of this paper is to review briefly the types of heaters most commonly found and point out some of the features of their installation and utilization which deserve more attention than is usually given.

Heaters using gas as fuel may be divided broadly into two classes:

First:—Central Heating:

In which one source of heat is used to supply several rooms and is seldom if ever in the occupied room.

Second:—Individual Room Heating:

In which the source of heat is a heating device within the room itself.

Note:—In this classification by "Source of Heat" is understood the device in which combustion is effected and not the means used for disseminating the heat.

CENTRAL HEATING:

Central Heating appliances are usually large, permanent structures intended for continuous service during certain seasons of the year, so located in the house as to give minimum annoyance from the dirt arising from the use of solid fuel, and well designed for convenience of operation. Because of these features gas,

* Member of the Committee on Utilization of Gas Appliances American Gas Institute.

although an ideal fuel, is forced to compete with other fuels on the basis of operating cost except in a few instances where cleanliness, convenience, or capacity are more essential.

A definite comparison between gas and coal on a fuel cost basis cannot be made to cover all cases as the qualities of gases and coals vary in different localities, but we may assume certain average values to serve as an example.

Let us assume for coal an average thermal value of 13,000 B.t.u. per pound.

A ton of 2240 pounds will have then 29,120,000 B.t.u.

When this coal is burned in a domestic furnace at least 40 per cent. is lost, so the available heat in a thousand cubic feet of gas is about the available heat in a ton of coal is about 17,500,000 B.t.u.

Assuming the cost of the coal to be $7.00 per ton we utilize for each $1.00, 2,500,000 B.t.u.

Let us assume for gas an average thermal value of 600 B.t.u. per cubic foot.

A thousand cubic feet will have then 600,000 B.t.u.

The efficiency of a well designed gas furnace is about 80 per cent. so the available heat in a thousand cubic feet of gas is about 480,000 B.t.u.

Assuming the cost of gas to be $1.00 per thousand cubic feet we utilize for each $1.00, 480,000 B.t.u. or about one-fifth as much as for a dollar's worth of coal.

Further we must consider:—

Sixty per cent. efficiency is rather high for the average domestic coal furnace, even when it is in continuous use.

No allowance has been made for the difference in labor cost. Gas requires no labor, coal requires intelligent labor for firing and careful attention must be given to the removal of ashes.

When these points are considered we find that the ratio of cost of gas to the cost of coal is materially lessened.

In a paper entitled "The Possibilities of House Heating by Artificial Gas as a Fuel," presented before the National Commercial Gas Association at Atlanta, Ga., in December, 1912, by A. F. Krippner of St. Louis, he says: ". . . with gas at 80 cents per 1,000 for the first 10,000 cubic feet and all excess at 50 cents per 1,000 monthly, it was found that the cost of gas fuel approximated twice that of hard coal at $8.00 per ton." I understand that many houses in St. Louis are now using gas as fuel in the furnaces. In other cities where the conditions are less favorable for gas, are found also occasional installations of gas fired furnaces.

So far we have only considered furnaces for continuous use as during the winter in the Northern States. If we pass to other seasons as early fall or late spring, or to the southern sections of the country, we find that the demand for heat is intermittent and the excess cost of gas over coal is vastly diminished or may even disappear. The reason for this is plain; the efficiency of the gas fired appliances does not decrease during intermittent use. The efficiency of the coal fired appliances, on the other hand, decreases rapidly as the rate of combustion drops below the normal and, obviously, if the fire is extinguished and relighted the operating cost is even greater.

So far as I am aware there is no entirely satisfactory combination furnace built in which either gas or coal may be used as the fuel. Several have been placed on the market but to each are objections that overbalance the supposed advantages. There are many satisfactory gas fired furnaces commercially available in sizes to suit all needs.

In the South a gas fired furnace alone may be installed but in the North, if cost of operation is the desideratum, a coal fired furnace for the continuous, and a gas fired furnace for the intermittent, use should be installed.

In the past the auxiliary gas fired furnace has not received the attention it deserves and as its utility is more and more appreciated its field should rapidly extend.

INDIVIDUAL ROOM HEATING

While gas burning Central Heating appliances are desirable under certain conditions it is in Individual Room Heating that the gas appliance is the undisputed leader, for the satisfactory heating result, obtained at a low cost, the convenience of operation and cleanliness, makes it the peer of all Individual Room Heaters.

Individual Room Heaters may be classified:—

In kind as,
- Stationary.
- Portable.

In character of heat as,
- Direct radiation from the flame.
- Indirect radiation from the flame.
- Convection.

In type of burner as,
- Illuminating flame.
- Atmospheric flame.

And may be further classified according to material of construction, kind of burner, method of connection, etc.

STATIONARY HEATERS

Are usually installed under mantels, whether or not a fire place for solid fuel exists. (A notable exception is the gas fired steam radiator which will be discussed in detail further on.)

PORTABLE HEATERS

May be of the true portable type, that is, having a flexible gas connection, readily detachable from the gas supply pipe, permitting an easy change of position during the season of use.

Or they may be of the semi-portable type, that is, connected in such a manner that during the season of use the position may be not readily changed, while during the season of disuse they may be disconnected for removal and storage.

CHARACTER OF HEAT

All gas heaters are so designed as to part with a minimum of heat by conduction, and so far as we know now all heaters transfer more heat by convection than by radiation, although there is a marked difference in heaters of different types of the proportions of heat thus transferred.

Part of the convected heat is obtained directly from the products of combustion and part by contact of the flowing air with the hot surfaces of the heater. In any case it is a maximum above the heater.

The radiant heat may be obtained directly from the flame or indirectly from those parts of the heater heated by the flame or products of combustion. That it is possible to control the distribution of radiant heat to a marked extent is shown by the accompanying curves, taken from the Report of the Committee on the Utilization of Gas Appliances to the American Gas Institute in October, 1912. When the radiant heat is direct the flame is luminous and when indirect it may be either luminous or atmospheric.

Luminous flames are of two kinds:

The round jet.

The flat flame, with either the sawed or slotted tip or the union jet tip.

The round jet is somewhat cheaper in first cost because it is merely a drilled hole, often in metal. The flat flame tip is of steatite, lava, or similar material and while somewhat more expen-

sive to make, is preferable to the round jet tip, as it is not so susceptible to variations of gas pressure and is also somewhat more efficient in general use because, due to its larger surface exposed to the air, the combustion is better.

Luminous flame heaters may be so constructed that no part of the heater is in front of the flame, in which case there is usually a reflector so designed as to reflect the radiant heat in the desired direction and the radiant heat is direct. Or they may be made so that the flames are entirely surrounded by the heater in which case the radiant heat is mostly indirect, a small part only being direct where the flames are visible through perforations in the enclosing body. With the latter type of heater the radiation is more symmetrical around the heater than with the former type which may be said to have a "front" from which the radiation is a maximum.

Because the radiant heat from atmospheric flames is relatively low these are employed only in heaters in which the radiation is indirect, that is, from a body which is heated by the flame. Many materials in many forms are used as radiators, the most common being asbestos, perforated sheet metal and ceramics. In this class may be placed also the gas fired steam or hot water radiators.

Having made this general classification we may now proceed to consider more in detail some of these types of heaters.

Fire place heaters, because of their position, should be preferably of such types as have the highest radiant efficiency, and with asymmetrical distribution of heat to prevent as far as possible conduction losses through the walls, although a few fire place heaters have been designed for a circulation of air in order to increase the proportion of convected heat. Flue connections are desirable for aiding in general ventilation but in no case should the full size flue for solid fuel be used, for so large a flue is unnecessary to carry off the products of combustion and a flue larger than needed will mean loss of heat. The small quantity of, and apparent innocuousness of the products of combustion of gas fire place heaters have led many builders to use what may be called "false flues," metal pipes three or four inches in diameter leading from the mantels within the partition walls and discharging, not through the roof, but into a loft or, even, in many cases only into the spaces between the studding. If a small metal pipe is well insulated it is unobjectionable though not so desirable as the tile flue built for solid fuel fires. Where for structural reasons it is impractical for it to go through the roof it may discharge into the loft, provided the latter is unoccupied and at all times well ventilated and a

proper cap or deflector is placed on the end of the flue. Discharging the products of combustion into a partition space, however, cannot be condemned too strongly and all should bend their efforts to the eradication of this practice, unsanitary because when the burner is improperly adjusted the flue products may be injurious to health, and unsafe because if the gas supply to the burner be increased above the quantity for which it is designed fire may result.

At this point it may be well to give some attention to the general subject of flues for gas heaters.

If the burner is for atmospheric flames a flue is to be recommended (and this applies to all gas heaters of whatever design) for the reason that when an atmospheric burner "flashes back" or burns within the mixing tube from the gas orifice instead of the burner end or ports imperfect combustion ensues, with consequent formation of carbon monoxide, a gas which is highly injurious to all animals. Heaters with such burners, therefore, except when of the smallest size, or when installed in well ventilated rooms where the occupants are actively engaged and in full possession of their faculties, should always be equipped with flues to carry off the products of combustion.

Another function of the flue, which has been already referred to, is to assist in the general ventilation of the room. This it does, not only by carrying off the products of combustion which otherwise would pass into the room, but the draft created by the heated products induces a flow of air beyond that which goes through the burner. High flues may defeat their purpose, especially if on the outside of the building because of the small volume and relatively low temperature of the products of combustion. Carbon dioxide being heavier than air must be kept warmer than the outside air, else it will not rise but remain in the flue, blanketing the flow.

During the combustion of gas the hydrogen, either free or as a hydrocarbon combines with the oxygen of the air to form water. With the average manufactured gas the quantity of water thus formed amounts to about forty pounds for each thousand cubic feet of gas burned. So if the products of combustion are cooled below 212 degrees Fahrenheit, the moisture will be perceptible and, even, a source of much annoyance.

This may explain the humidity often noticeable in rooms where the gas heaters are not flue connected.

This formation of water vapor occurs whenever gas is burned regardless of the kind of burner, and precautions should be taken

either to keep the products at a temperature above the condensing point or to provide means for disposing of the condensed water whenever gas is burned in such a quantity that the condensation may be an annoyance. When gas room heaters are used as auxiliaries to other sources of heat and burned for short periods, the water vapor formed is generally not sufficient to cause annoyance.

COMMON TYPES OF HEATERS:

The enormous demand for gas heaters has naturally caused great activity among inventors, and devices almost without number have resulted. Use has proved the value of these and eliminated, for common use, all but a relatively small number of types for which distinctive names have been adopted by common consent. Heaters of the same type by different manufacturers are usually quite similar in appearance and heating effect, differing only in details of design.

Among the most common types are:—

Reflector heaters (Fig. 1), having luminous flames and polished copper backs to reflect both the heat and the light. The burners are usually near the top of the body, a sheet metal box open at the front, preferably provided with a screen to prevent inflammable materials from contact with the flames. These heaters, which are made in many sizes, are suitable for nearly every purpose and owing to the simplicity of operation, ready portability, and potent and cheerful heat they are much in demand for residence and, to a somewhat smaller extent, more public places. The distribution of heat is asymmetrical, being materially greater in front than at the sides and back. (See Chart, Fig. 1a.)

Radiator heaters (Fig. 2) usually have luminous flames below one or more sheet metal tubes. If but one tube is used it is cylindrical, about eight inches in diameter and fifteen inches high with several burners inside and near the bottom. If several tubes are used they may be cylindrical or elliptical in section, about four inches in greatest diameter and from twenty-four to thirty inches high. There is usually one flame inside of, and near the bottom of each tube. Modifications of this type provide for atmospheric burners, and also for concentric tubes, so arranged that a longer passage is provided for the heated products and for circulation of air to increase the heat of convection.

In this type of heater the tubes must not be so much closed that there is not at all times a free escape for the products of combustion. These heaters are made in many sizes, and are readily

portable. The distribution of heat is nearly symmetrical, being maximum at the top and greater at the sides than the ends, in proportion to the projected areas. (See Chart, Fig. 2a.)

A heater which is somewhat similar to the proceeding has an atmospheric burner below a cone of perforated sheet metal which may be wholly enclosed in a sheet metal case or the case may extend only on three sides of the cone which is then in plain view from the front.

Asbestos back heaters (Fig. 3) have a vertical sheet of asbestos board on which is attached, by means of silicate of soda, long fibre asbestos. Immediately below and in front is placed an atmospheric pipe burner, the flames from which heat the fibres to incandescence or at least a bright glow.

In a modification of this the back is a box of sheet metal, the front of which is perforated and similarly covered with asbestos fibre. The box is the gas chamber, the flames burning from the perforations and heating the fibre.

The appearance of either of these heaters when lighted is quite agreeable but as the asbestos stains quickly the appearance when unlighted is not attractive.

A serious disadvantage of the heater with the perforated back is that because of the insufficient air supply to the upper ports, which lie in the path of the products of combustion from the lower ports, there is decidedly poor combustion.

In England many heaters are made having a row of atmospheric burners, over each of which is supported a suitable body of refractory material. These bodies are usually of fire clay, or similar material which becomes red- or white-hot when heated. A similar heater employs spirals of silica glass supported inside of silica glass chimneys. Both are efficient as heaters and not unpleasing when not lighted. (See Chart, Fig. 3a.)

All of these heaters may be portable and when intended to be so used, should be so designed that the temperature underneath will not be sufficiently high to damage the floor or its covering.

These heaters, either of the portable type or preferably modified to suit the location may be installed in fire places.

Where natural gas is available, perforated, hollow balls of fire clay of different designs are placed on a flat burner in an ordinary coal grate, but this type of heater is not desirable with manufactured gas.

Another heater much used in fire places is the gas log (Fig. 4). This is usually of ceramic material, the outside finished in imitation

of fire wood. An air mixer is provided below the log which is hollow, the interior space being used as the mixing chamber. The gas burns from perforations in the log and a lively effect is provided by patches of asbestos fibre attached to the log to glow when heated. While the combustion with the gas log is not perfect because of the insufficient secondary air, it is a fairly efficient heater (See Chart, Fig. 4a) and because of its appearance it enjoys a deserved popularity.

Gas fired steam radiators (Fig. 5) are now much in vogue. These are similar in appearance to the common steam radiator, made in any desired number of sections and with an atmospheric burner beneath. A small quantity of water placed in the radiator is heated by the burner and the upper part of the radiator serves as the steam chamber. A regulator operated by the steam pressure controls the gas supply and maintains a nearly uniform pressure.

These radiators may be connected in the regular low pressure steam heating system and gas heated as an auxiliary at times when the regular steam supply is cut off. For buildings, a large part of which may be unoccupied at times, this feature is desirable for it is then unnecessary to operate the central plant to supply heat for only a few rooms.

Because of the weight of this heater it cannot be readily moved so is of the stationary or semi-portable type. (See Chart, Fig. 5a.)

A heating system is now being placed on the market the units of which are similar in appearance to steam radiators, each having an atmospheric gas burner in a combustion chamber in place of the central sections. A suction fan at a convenient point, connected to the various radiators with wrought iron pipe draws the products of combustion through the sections which are so proportioned as to absorb most of the heat. All of the air for combustion is taken in as primary air and as the products are positively discharged there is a marked ventilating effect.

Power to operate the fan is needed, however, and provision must be made for disposing of the water condensed from the products of combustion. As solid pipe connections are required for the exhaust line this heater must be considered stationary.

I make but passing reference to heaters with "surface combustion" burners as these are still in but the experimental stage. Surface combustion burners are a development of atmospheric burners in which all the air for combustion is primary and "flashing back" is prevented by the use, at the burner orifices, of refractory material loosely placed thereon. The function of this material is

to provide a gradually increasing area of the burner ports towards the point at which combustion is effected. The result is that backwards from this point the velocity of the stream of gas and air is more rapid so that at a point just inside of the surface the velocity of flow is greater than the velocity of propagation of explosion, and combustion anterior to this point is prevented. As gas and air are supplied in proper proportions for perfect combustion, the combustion is effected almost instantaneously with a minimum of flame and high intensity of heat development. Such burners, at the present time, require a blower to supply the primary air.

For stationary and in many cases for semi-portable heaters, solid metal pipe connections may be made for the gas supply and it is obvious that such connections are the best in every way. This method of connecting cannot, however, be used with portable heaters which from their nature demand a connection sufficiently flexible to permit limited movement when in use and ready connection or disconnection, without use of tools, from the permanent gas supply fitting.

Flexible tubing of various materials for this purpose may be classified.

Rubber Tube:—

No kind of rubber tube is satisfactory as the illuminants commonly found in manufactured gas combine with the rubber and soon render it unfit for use.

Composition Tube:—

A mixture of glue, glycerine and lithage (or their equivalents) is used as a gas proof coating on a tubular base, usually of braided cotton, which may or may not be stiffened by means of a helically wound wire. When carefully made this tube is fairly satisfactory but if the proportion of glycerine is insufficient the tube will stiffen quickly while if it is excessive the composition becomes so hygroscopic that during humid weather, water will collect on the tube.

Metallic Tube:—

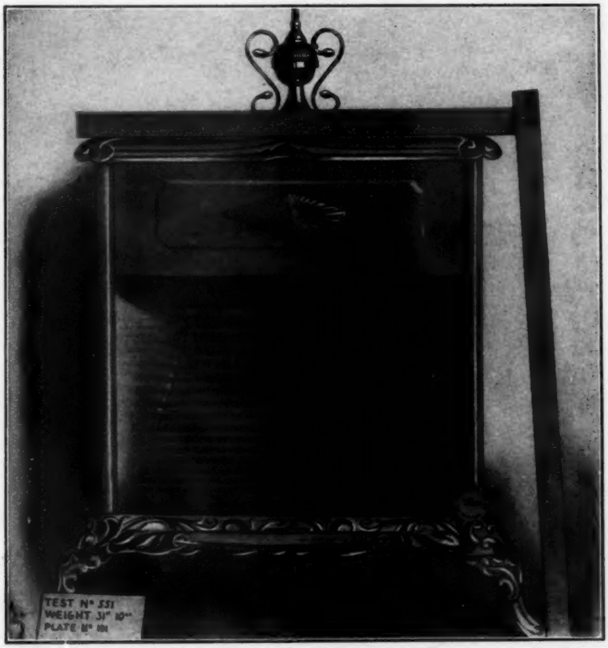
This is made of helically wound strip metal of special section so that the contiguous turns are interlocked and made gas tight by a gasket of rubber or asbestos thread. The chief objection to this tubing is that if twisted in a direction opposite to the winding, the turns will open sufficiently to permit the leakage of gas. If carefully used, however, it is a safe and satisfactory flexible gas conduit.

Flexible tubing of any kind must be so made as to permit its ready connection or disconnection from the permanent gas supply

fittings. By far the most common means is the rubber "slip-end," a cylinder of elastic rubber, permanently attached to the tube, the axial hole through it for the passage of gas being of the proper size to fit over the metal "hose end" of the fixture to which it is attached. This end depends on the elasticity of the rubber for its holding power and as with time the rubber hardens the value of the end decreases. Frequent attachment or detachment also wears the rubber so that in time it becomes too large to properly hold. In spite of these disadvantages it is so universally used and its shortcomings are so well known that its ready applicability make it a satisfactory device.

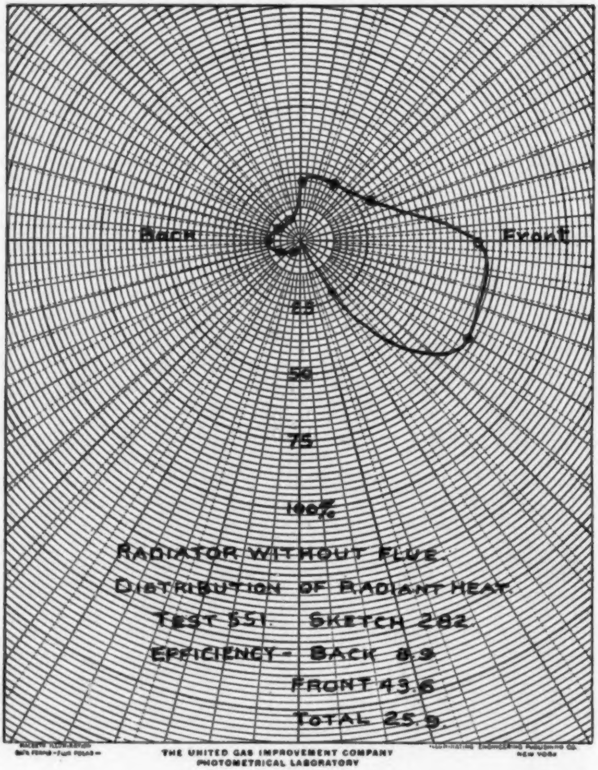
Several substitutes for the rubber slip end are now available, the most practical being a "union" which by means of a wing nut may be tightened or loosened without the use of tools. This is so decidedly an improvement over the slip end that its more general use is very probable.

I shall say no more regarding the hygienic effect of gas heating, because its principles are similar to heaters using other fuels, than to call attention to the real sterilization of air effected by its passage through the flame. Many interesting tests to prove this fact have been made, especially in England, details of which may be found in the Gas Trade Journals, particularly the "London Journal of Gas Lighting," during the past two years.



Radiator Heater

Fig. 1



Reflector Heater Chart

Fig. 1a

Note:—Curves illustrating the distribution of radiant energy were made with a calorimeter, which consisted of a lune revolving around the heater, the axis of the lune being approximately on the longitudinal axis of the heater.

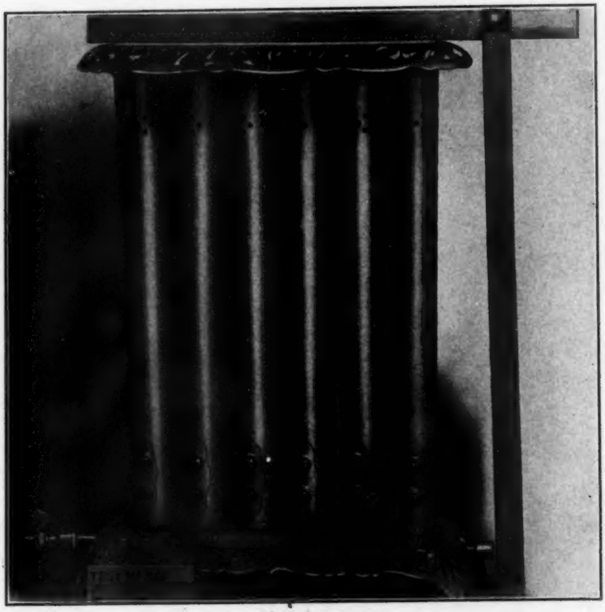
Readings were made directly above the heater 0°, then at 30°, 60°, 90°, (horizontally opposite the center of the heater) 120° and 150°. No reading was made at 180°, or directly below the heater, because of the difficulty of properly supporting the heater in order to make such a reading, and because it was found by experiment that this reading could be assumed to be 0°.

These readings were made at both the front and back of the heater, the mean of the readings at 30°, 60°, 90°, 120° and 150° being assumed the efficiency for the front or the back of the heater, as the case may be.

The total efficiency is the mean of all the readings around the heater, or 0°, 30°, 60°, 90°, 120°, 150°, 210°, 240°, 270°, 300° and 330°.

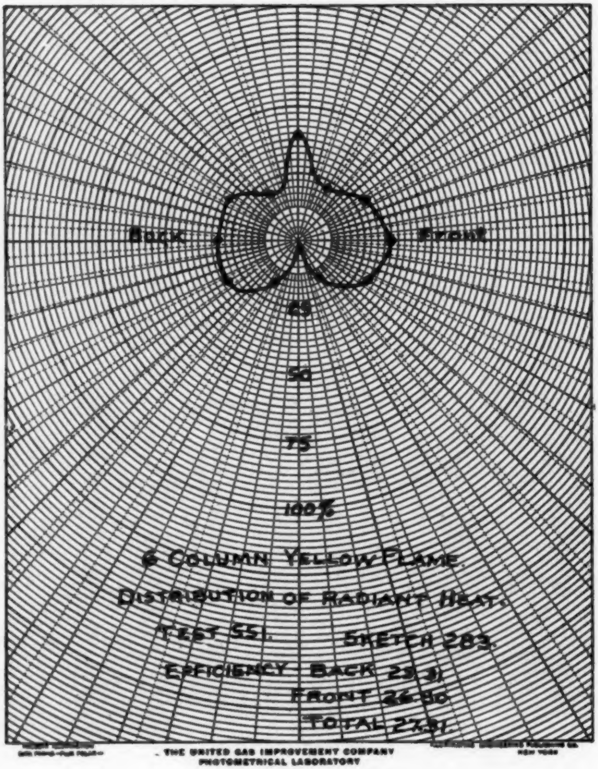
It is to be noted that these curves were made for the purpose of showing the *distribution* of the radiant energy only.

The difference between 100 and the total radiant energy may be assumed to be the energy dissipated in the form of heat of conduction or the heat of convection.



Reflector Heater

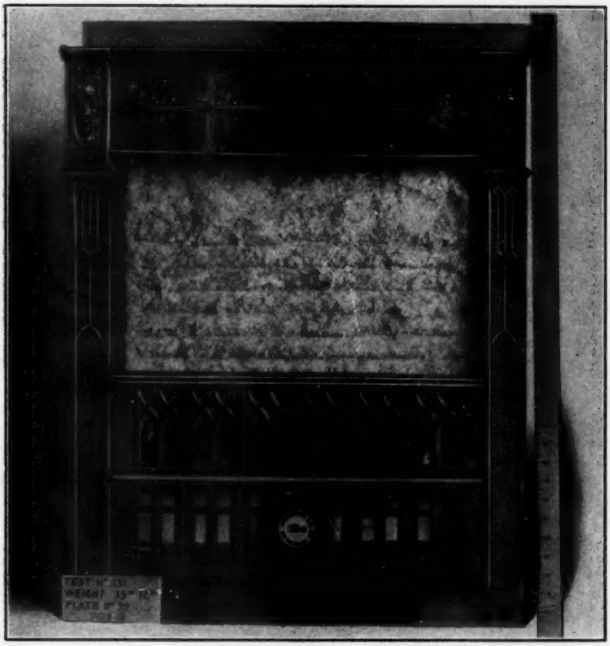
Fig. 2



Radiator Heater Chart

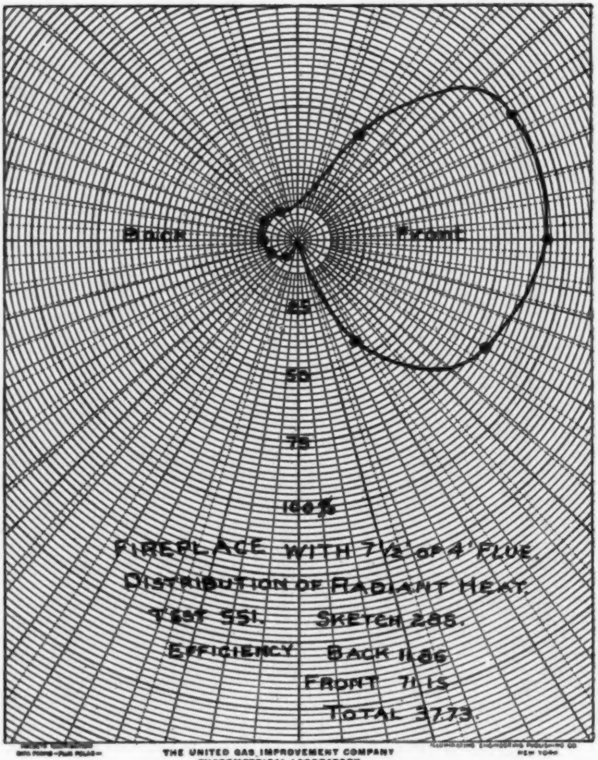
Fig. 2a

See Note under Fig. 1a



Asbestos Fire Back

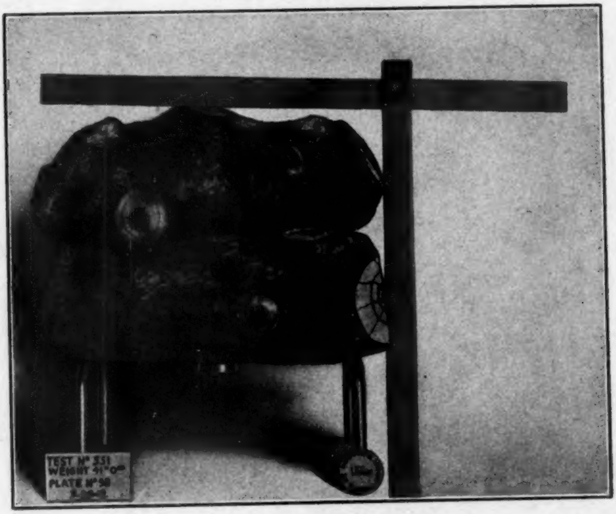
Fig. 3



Asbestos Fire Back Chart

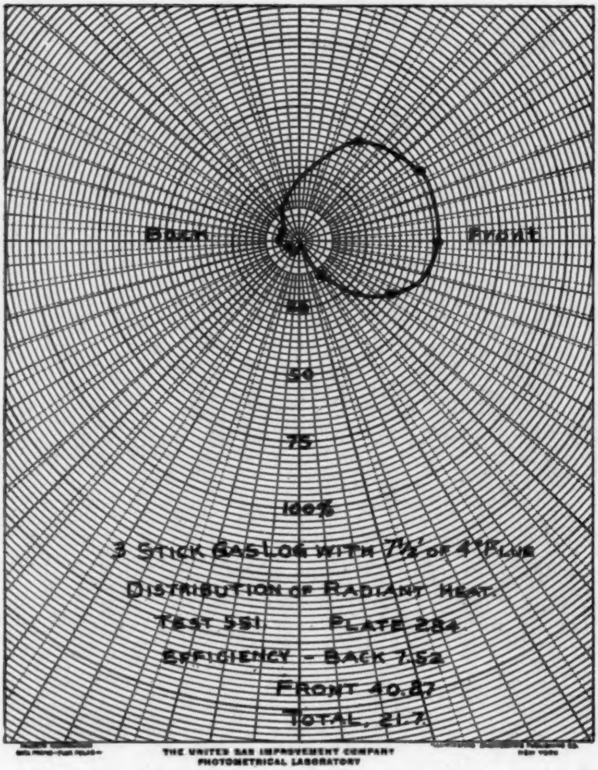
Fig. 3a

See Note under Fig. 1a



Gas Log

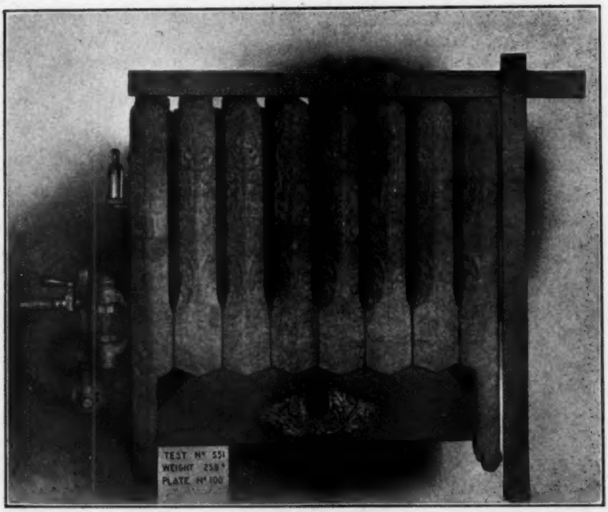
Fig. 4



Gas Log Chart

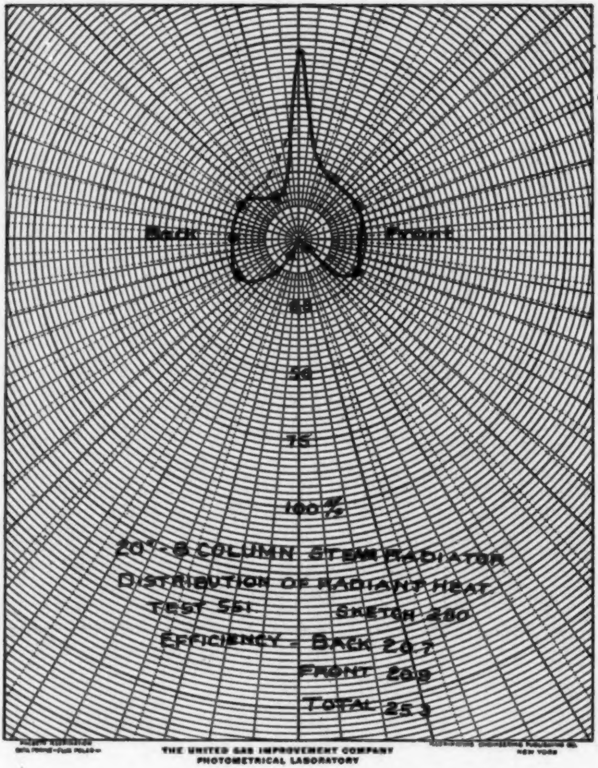
Fig. 4a

See Note under Fig. 1a



Gas Steam Radiator

Fig. 5



Gas Steam Radiator Chart

Fig. 5a

See Note under Fig. 1a

Synopsis of the paper

The paper deals with gas heating appliances of various types.

Explains the relative costs of heating by gas and coal.

Treats of the phases of central heating and individual room heating. Explains the value of the different ways of connecting the gas to radiators.

Illustrates a number of heaters with a chart to each showing their efficiency by direct radiation and by heat of convection.

DISCUSSION

A Member: I feel, that in making comparisons between gas and coal heating, we often lose sight of the conditions of operating of gas heaters as against coal fired heater, for instance, in burning coal. If your room becomes too warm you do not reduce the coal consumption. If you open check draft you decrease the efficiency of the heater but do not decrease the coal consumption. With gas radiator you simply turn off the gas if the room gets too warm or lower it and you decrease the consumption of gas.

I call to mind an office about 15 x 45 ft. with separating walls where they formerly used about three-quarters of a ton of coal per month in heating. This winter in November they used a trifle less than $2.00 worth of gas. In December which you know is a very cold month, the gas was a little less than $4.00. It is heated with three heaters, two of which are turned off when they leave the office at night. If lighted in the morning they get steam up in a few minutes, against three-quarters of an hour in a coal heater.

For churches or offices, restaurants, stores and buildings of like nature, under certain conditions gas can be used as cheaply as coal. It does not take nearly as long in the morning to get steam up and you can shut it off and decrease, or cut off immediately, your gas consumption, which you cannot do with a coal fired furnace.

Mr. Blaney: The discussion of gas for heating is quite interesting, but one point Mr. Barrows did not cover and that is relative to the insurance.

Mr. H. J. Barron: In regard to the question of insurance companies rates, I think if the gas people were more enterprising, this difficulty would be over come. The trouble is not with the engineers, whose devices are practical and up to date, the whole

trouble is with the management of the gas companies. Everybody I think knows that there have been more fortunes made in gas exploitations than almost any other line of effort in this country, it is a question of exploitation by the capitalists who are interested in gas. That is why the gas companies are up against the problem of municipal ownership of which they are always complaining, yet they are using every effort to combine with the plants generating electricity and endeavoring to make the public believe that the price is reasonable. Engineers who are in control of the manufacture of gas appliances know very well that gas could be produced profitably and pay good dividends generally on 50 cent gas, where they now want 80 cents to $1.00. Heating devices utilizing gas would have ten times the demand they have if the gas companies were broadly managed. I don't mean to say they are not ably managed for money making. They are probably ably managed because they have made more money than most any other business. Their propaganda at present in having fine stores and able men going around advising the use of gas appliances is foolish. They would have ten times the demand if the management was more ready to appreciate the public view. The public believe they are being robbed in the high price of gas.

Mr. Chapman: This paper very interestingly explains the use of gas for fuel, but I would like to ask in what way he obtained a greater efficiency in the steam gas radiator than in the ordinary gas tube radiator.

Prof. Kent: I wish the author had explained more fully the method of determining the efficiency of these radiators.

Mr. Barrows: Regarding the question of insurance the Gasteam radiator has been approved by the National Underwriters Laboratory at Chicago. What local underwriters' boards do, I don't know. Why its use should be considered an extra hazard is surprising to me. In some cities it is not considered as extra hazardous and no additional premium is charged. The floor temperature of a well designed gas steam radiator is low even after continuous burning after several hours and the burners are so protected that it is impossible for any loose material to come in contact with the flame.

Referring to the efficiencies of the heaters shown on the curves, I feel that I should again call attention to the fact that these curves only show the relative distribution of radiant heat, and do not show in any way the total efficiency of the heater, or the absolute radiant efficiency.

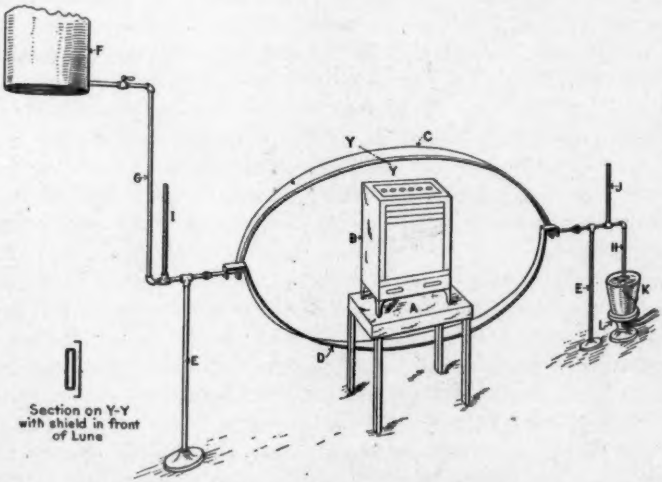
Any gas heater with a burner so designed that perfect combustion will ensue, will be 100 per cent. efficient, if it is not flue connected.

By this, I mean that all of the heat which is generated by the combustion of the gas, will be available for the heating of the room in which the heater is placed, this heating being effected by radiation or convection (as practically every gas heater is so supported on legs that the transference of heat by conduction need not be considered).

Some heaters are more efficient than others, not because they are better heaters, but because the character of the heat is somewhat pleasanter, and it was for the purpose of determining the characteristics that produced the more desirable results, that we started our study of the distribution of radiant heat, intending to supplement these tests by others which would study the convection currents.

Unfortunately, other work of more pressing nature has intervened, and it has been impossible for us to continue our investigation.

To measure this heat, a special form of calorimeter was devised, and a diagrammatic sketch of this piece of apparatus is appended.



A wooden stand, "A," supports the heater, "B."

Revolving about the heater (except underneath) there is a hollow lune of copper, "C," about 3 x ½ inch in section.

The side of the lune towards the heater is blackened, in order to absorb as large a proportion as possible of the radiant energy which falls on it.

A shield, "D," of bright tin is arranged on the same axes as the axes of the lune, and is used for shielding the lune from the radiation as will be described further on.

The lune and its shield are supported on the stands, "E," by means of the pipe axes, "G" and "H." "G" leads from the water supply "F," and "H" is an outlet to a bucket, "K," on the pan of a scale, "L." The thermometers, "I" and "J," give the temperatures of the inlet and outlet water.

To operate, the heater is lighted and allowed to burn until it has reached a uniform temperature. Water is then allowed to flow from the tank through the lune. The lune is set in any desired position above the heater, horizontally opposite the center of it, or at any angle, and the rise in temperature of the water through the lune is observed by means of the two thermometers, the water being weighed to determine the number of heat units absorbed with that particular position of the lune.

Knowing the weight of water which passes through the heating element and its rise in temperature, we may, of course, determine the number of B.t.u.'s absorbed.

From simultaneous observation of the rate of combustion of the burners in the heater, we know the amount of energy generated by the combustion, and in this way we may determine definitely the heating efficiency at any one point at which the heating element may be located.

The test was not conducted in this way, however, as we only wished to determine the relative distribution of the radiant energy. The heating element was placed in one position and an observation made, the rise in temperature of the water being due, then, to all of the heat to which the heating element was subjected, both heat of convection and heat of radiation. A polished shield was then interposed between the source of heat and the heating element, and another observation made, the rise in temperature in this case being due only to the heat of convection, as the heat of radiation was cut off by means of the shield.

The difference, then, is the amount of heat due to radiation.

The curves were obtained by plotting as the radii vectors the percentages of heat received by the heating element in the form of radiant energy.

Mr. Hart: Does the gas steam radiator give more heat per cubic foot of gas than the direct burner?

Mr. Barrows: It does not give any *more* heat. Why the gas steam radiator is as popular as it is is somewhat of a question. Some of the arguments for the advantage of the gas heater are nullified by the gas steam heater. In a gas steam radiator about fifteen minutes are required to get up steam, that is, for the first fifteen minutes after the heater is lighted, it is storing up heat faster than it is giving it off and then after turning the gas out, it continues to give off heat for fifteen minutes. I don't know why they are as popular as they are, but they certainly are as efficient as any other heater in common use. Any gas burning device placed in a room without flue connection, must be 100 per cent. efficient if the burner is burning properly, that is, we are getting out of the gas all the heat that is in it. Whether we are utilizing that heat to the best advantage, we don't know and the investigation that we started to make on the distribution of radiant heat was for the purpose of endeavoring to determine the efficiency of the heaters, not as gas burning appliances, but as heaters.

General opinion in England seems to be that radiant heat is more desirable than heat of convection. We don't know very much about it and are anxious to have you people who are so thoroughly in touch with heating problems give us the benefit of your experience on this point and tell us what we ought to do.

We have been building heaters along lines that have not changed very much in the past fifteen or twenty years. We think they can be greatly improved and we hope the heating engineers will tell us what we should do.

Dr. Franklin: There is another viewpoint that has not been considered. A difference of opinion exists as to the relative desirability of radiant and convection heats, which is this: Convection heat is more or less uncontrollable; the air coming in contact with the surface is warmed and then wanders throughout the room in a more or less lawless fashion. This air could be controlled in a measure by deflectors, blowers and other devices but in general, it is uncontrollable. On the other hand, radiant is controllable. It is more desirable to have a heating surface from which a larger amount of heat is given off by radiation, the idea being that objects in the room can be directly warmed if the radiation heat is in large quantity as compared with the convection heat. This is accomplished by suitable designing of the surface of the heater.

The excuse for the steam radiator is that a very large surface can be had at a given temperature with a minimum expenditure

of energy. In case of direct heating surface, we are limited more or less. The same result as with steam would be gotten with a large enough sheet iron surface evenly heated. A large quantity of the steam is heated and kept distributed so that the amount of radiated heat is relatively great. In this way the heat in the different localities of the room is under control as compared with the more haphazard way of heating by convection.

A Member: Another reason for the gas steam radiator is that it is especially suited for stores where people are likely to come in contact with the heater. The temperature on the surface of the heater is about 215 deg. against which persons would not get burned. The ordinary tubular heater has nearly twice that temperature.

Mr. Palmer: Think perhaps another reason for the gas steam radiator is the possibility of control. With any other form of gas radiator you have no chance for automatic control. Also in the matter of lower temperature, if all heaters as Mr. Barrows said are 100 per cent. efficient, I think we want surface of lower temperature and more of it rather than surfaces of high temperature that are obtained in most gas burning heaters.

CCCLXXIX

CAPACITY OF STEAM PIPES AT DIFFERENT PRESSURES

BY JAMES S. OTIS, MEMBER

That we may understand the theory and comprehend the many different stages in the development of a formula for the theoretical and actual flow of steam through pipes of varying sizes and lengths, it will be necessary to give a few definitions and make some explanations of the properties of the substances under consideration.

FLUIDS AND SOLIDS

Every substance in nature is either a fluid or a solid, and the distinguishing features of each are sometimes very confusing to casual observation. The status of form or degree of hardness is very misleading, since a substance may be very hard and yet be classed as a liquid, while on the other hand, it may be very soft and be classed as a solid, the distinguishing feature being its resistance to a change of form.

When a continuous alteration of form of a substance is produced only by a stress exceeding a certain value it is called a solid, however soft and pliable it may be; but when the smallest stress, if continued long enough, will cause a perceptible and increasing change of form the substance is called a liquid, however hard it may be. Thus a piece of tallow is a solid, while a stick of sealing wax is a liquid.

VISCOSITY

All fluids are more or less viscous: That is, they will resist a *sudden* change of form, yet the slightest stress will, in time, change any formation. A block of pitch will resist great sudden pres-

sure, and even if hit with a hammer will chip and break into splinters, yet the mere weight of its parts, if left long enough, will cause it to flatten and finally flow like water.

LIQUIDS AND GASES

Any region of space enclosed by a rigid boundary can be filled with a liquid which then takes the form and shape of the boundary. Remove any portion of the fluid and if it be a liquid a portion of the space will be empty; if it be a gas the remaining portion will immediately expand and fill the space. Therefore, if it takes a definite quantity of a liquid to fill a certain space, while a quantity of gas, however small, will expand to entirely fill the space, the simplest definition would probably be:—

A solid has volume and shape.

A liquid has volume, but no shape.

A gas has neither volume or shape.

VAPOR

It is very essential that we distinguish between a gas and a vapor; for, while they have some properties in common, they differ materially in properties most essential to our considerations; vapor, for instance, being readily condensed to a liquid either by increasing its pressure or lowering its temperature, while a gas very strenuously resists this change of form. The Ideal or Permanent Gas, so often mentioned by the earlier writers, was supposed to entirely resist this change. In the light of modern experiment, however, there is no Permanent Gas, for, with but one or two exceptions, they have all been reduced to a liquid; their resistance to this change of form depending merely on pressure and the critical temperature.

CRITICAL TEMPERATURE

The critical temperature is that point above which the space will be filled with a homogenous mass; and below which it will separate into two phases. This ranges from approximately absolute zero (—459.4 deg. F.) to a very high temperature.

In gases the critical temperature descends and in a few instances it is so low that the critical point has not yet been reached although it is conceded by all scientists that could absolute zero be attained it would pass their critical point.

The vapors take the ascending scale and, as in the case of water vapor, require a very high degree of heat to reach their critical

temperatures. If a quantity of water be placed in a closed vessel having some free space and heat be applied vapor will fill the space and the water will remain at the bottom. As the heat increases the vapor will become more dense and the pressure on the vessel very rapidly increase.

When the temperature in the vessel reaches 362 deg. C. or 683.6 F. the remaining water will unite with the vapor and form one homogenous mass. If more heat be applied the mass retains this same form but the pressure increases, not in the proportion of saturated steam, but as that of a gas. Therefore 683.6 F. is called the Critical Temperature of steam, and above that temperature it is supposed that its properties are those of a perfect gas.

WATER

Strictly speaking, water is the oxide of hydrogen, but in popular use it is a name applied to a great variety of different substances. However, as these contaminating substances amount to only about one-tenth of one per cent. by weight of the water and as they do not materially effect the general result we are seeking we will consider water as a pure liquid, having many peculiar properties not found in other fluids. For instance,—it attains its greatest density at 39.2 F. and slightly increases its volume as heat is applied or withdrawn and very materially expands when freezing; all other liquids decreasing in volume as heat is withdrawn. It is only slightly compressible, and at a stationary temperature its compressibility up to 1,000 pounds per sq. in. is in direct proportion to the weight applied. But again differing from all other liquids its compressibility decreases as its temperature increases.

SPECIFIC HEAT

All substances will absorb heat, and the readiness which they show for this absorption is termed their specific heat. Water very readily takes in heat and by common consent and practice it has become the standard of measurement and its specific heat established as 1; all other liquids, except methyl alcohol, having a specific heat of less than one.

LATENT HEAT OF STEAM

While water very readily absorbs heat, it very strenuously resists a change of form, and when changing from a liquid to a

vapor without increasing its temperature, it requires almost a thousand times as much heat as it would to increase its temperature as water one degree. This absorption of heat is known as the Latent Heat of Steam.

STEAM

Steam is water vapor. As such it retains many of the properties of water and takes some of the properties of a gas, approaching a perfect gas as its temperature increases.

SATURATED STEAM

When quantities of heat are applied to water some of it changes its form and becomes a vapor. If this vapor is confined in a closed space, as in a steam boiler, and the point of equilibrium is reached, that is the point where the water will refuse to give up any more vapor at that temperature, the space is said to be saturated with vapor, and the density of the vapor is then the maximum which can exist in the presence of the water at that temperature. The pressure at that temperature is called the saturated pressure. Hence *Saturated Steam is steam in contact with water at a temperature which is the boiling point of the water and the condensing point of the steam.*

SUPERHEATED STEAM

Superheated Steam is steam out of contact with water and heated above the saturated temperature. This steam, as in saturated steam, increases in pressure or volume as heat is applied but not in the same ratio, following more closely the rules of gas expansion.

DRY STEAM

Dry steam is saturated steam free from mechanically mixed portions of water.

WET STEAM

Wet steam contains water in suspension.

Suppose we have a cylinder closed at one end and fitted with a piston that will exert a constant pressure (p) during the entire travel upward. Suppose we place a certain quantity of water (one pound = w) at any temperature (t) above 39.2 deg. and let the surface of the piston rest upon the surface of the water, at the constant pressure (p). If heat be now applied to the bottom of

the cylinder the temperature of the water will gradually rise until a certain temperature is reached. The piston will rise slightly but still cling to the surface of the water. But at this point of temperature (t), the value of (t) depending upon the pressure (p) exerted by the piston, steam will begin to form and the piston will leave the surface of the water. As steam is formed more heat is taken in and the piston rises in a definite ratio to the amount of water evaporated. As the pressure (p) remains constant the temperature (t) of the steam and water will remain constant until all water is evaporated. During this process, so long as there is any water in the cylinder, the steam is said to be saturated. If after the water is evaporated heat be still applied the temperature will increase and the piston rise, but not in the same ratio as before, and the steam is then said to be superheated. If water of the same temperature be introduced, some of it will be evaporated, and if any remains in the cylinder the steam again becomes saturated steam. If now a heavier weight be applied to the piston while the temperature remains the same, small globules of water will form in the steam and it becomes wet. These will quickly fall, and the latent heat that had changed this water into steam will be used to raise both the temperature of the water and the steam to a point of equilibrium and the steam again becomes saturated. If, on the other hand, the weight of the piston be made lighter and no heat be added, the steam will expand, drop in temperature, and a portion of the water will be evaporated into steam; therefore there is always a definite ratio between saturated steam and its water. That is, the product of its pressure and density is in direct relation to its temperature, and is a constant.

ISOTHERMAL EXPANSION

When water, by the application of heat, is converted into steam at a constant pressure its expansion is along Isothermal lines; consequently, if it is afterwards expanded or contracted evaporation of condensation must accompany the process.

ADIABATIC EXPANSION

If steam initially dry be allowed to expand adiabatically (namely, without taking in or giving out any heat) it becomes wet. A part of the steam is condensed by the process of adiabatic expansion, at first, in the form of minute particles suspended throughout the mass. The temperature and pressure fall; and as that part

of the substance which remains uncondensed is saturated, the relation of pressure to temperature throughout the expansion is that which holds for saturated steam.

ABSOLUTE AND INDICATED PRESSURE

If any perfectly sealed vessel containing water and saturated steam, be allowed to cool, the steam will condense and the space it occupied become a *vacuum,* or nearly so, and the weight of the atmosphere then exerts a pressure upon the outside of the vessel of 14.697 lbs. per sq. in. If heat be again applied and steam formed it will not neutralize this pressure until the water reaches the temperature of 212 deg. F. The water will boil or give off a vapor in this perfect vacuum at 32 deg. and completely fill the space, but this vapor will not accumulate sufficient pressure to overcome the atmospheric influence until the water reaches 212 deg. and the steam pressure is 14.697 lbs. per sq. in. As a perfect vacuum is not obtainable in actual practice, manufacturers have arranged their Steam Gauges to indicate from this neutralizing point (generally given as 14.7 lbs.) calling that pressure O, and all pressures below this point are indicated as vacuum, and all above this point as pressure. Hence, Indicated Pressure means that shown by the gauge, while Absolute Pressure is the lowest temperature at which water will give up a vapor in a perfect vacuum.

PROPERTIES OF STEAM

The relations of *pressure, volume, temperature* and *density* of saturated steam received considerable attention from the earlier philosophers, especially from Regnault, and elaborate tables were prepared, giving the relative value of these properties. Afterwards H. L. Callender modified and corrected the tables by using two constants instead of one, and in the main these were correct; but slight errors of calculation had crept in and Dr. Mollier of Dresden was the first, we believe, to correct the calculations and present them in a form that could be considered reliable. These tables have been revised and transcribed by Ewing from the metric system of measurement and the Centigrade readings of temperature to the foot measurement and the gradings of the Fahrenheit thermometer, and we have accepted these tables as our authority in making our calculations.

In order to determine the requirements of our problem let us suppose that we have a certain pipe (d) inches in diameter and

(L) feet in length through which we desire to pass a certain quantity (w = weight in pounds) of steam. Suppose we have a boiler developing a certain constant pressure (p) at one end and a condenser (C) at the other end.

So long as there is no condensation at C there evidently will be no flow of steam through the pipe (d), regardless of the pressure at (p).

When condensation starts at C steam will commence to flow through (d) at a velocity depending entirely on the sectional area of the pipe and the condensation at C. Thus showing that the *velocity flow is independent of the initial pressure,* modified of course by the law of continuity.

If the condensation at C is such as to induce any considerable velocity of flow in (d) there is always less outward pressure at C than at (p); and as the condensation increases the difference in pressure increases until the maximum carrying capacity of (d) has been reached; the volume (w) delivered depending on the initial pressure at (p) to sustain the steam at saturation during the entire distance of its travel. If, however, the pressure at C becomes less than 57.7 per cent. of the pressure at (p) the volume flow will not increase, regardless of any further drop in pressure.

If there be less pressure at C than at (p) the temperature of the steam is less and the volume per pound has increased. In as much as the area of (d) is constant it is evident there must either be an increased velocity or a condensation of steam, or both.

Suppose there is no loss of heat from exposure of the pipe, and no acceleration from the forces of gravity (the pipe being horizontal), then the loss of heat must be due either to friction or the development of kinetic energy to push the mass along at the increased velocity, or both.

As the loss by friction must be along Isothermal lines there must be condensation to correspond and a consequent drop in temperature; but by examining the steam tables we note that the latent heat is less in the higher temperature than in the lower, and consequently there has been a greater increase in volume than the condensation corresponding to the higher temperature would produce. This excess of volume must necessarily require some heat to give it the required increased velocity.

Under normal conditions there is a very considerable loss of heat from the exposure of the pipe. This is also Isothermal and produces condensation, with a corresponding increase in velocity due to the difference in latent heat requirements. As this loss is dead, the kinetic energy must come from the remaining live steam. There-

fore in determining the amount of steam delivered at (p) or rather the amount delivered at C, if the first quantity is fixed, we must consider the problem under three phases:

First: The amount of heat lost by friction in passing through the pipe.

Second: The loss of heat from the exposure of the pipe.

Third: The amount used as kinetic energy to drive the increased volume at a higher velocity.

The loss of heat from the exposure of the pipe can be treated as a constant, for while it varies slightly from the drop in temperature, as the pipe reaches its terminal, this difference is so slight as to be negligible in ordinary calculations.

The most carefully conducted experiments along this line have been made by the Navy Department of the U. S. Government and the Armstrong Cork Co. at their experimental station near Pittsburg. As we must adopt some material as a standard we have accepted their Standard Nonpareil.

Let us now summarize the general laws that must be considered in the solution of our problem.

1. The pressure at any point of a plane in the interior of a fluid is the intensity of the normal thrust estimated per unit of the plane. (Bernoulli.)

2. There is an equality of fluid pressure in all directions, modified only by gravity and viscosity. (Pitot.)

(Theorem:—In a fluid at rest under gravity the pressure is the same at any two points in the same horizontal plane.)

3. The Law of Continuity: In any pipe interior, completely and continuously filled with liquid, the inflow must equal the outflow. This, in the case of a compressible fluid, would mean that the weight flowing out must equal the weight flowing in.

4. That the surface friction in a pipe is in inverse proportion to its size. (Mariotti.)

5. That with fluids in motion the distribution of pressure is the same as with fluids at rest. (Bernoulli.)

6. That the frictional resistance is independent of the pressure between the fluids and the solid against which it flows.

7. The volume of a given mass of gas varies inversely as the pressure, provided the temperature remains constant. (Boyle's Law of Perfect Gases.)

8. The temperature of saturated steam depends on the pressure. (Regnault.)

As steam is another form of water it naturally retains many of its properties: Yet, as a compressible fluid, it takes some of

the qualities of a gas, therefore it will be necessary, in the consideration of our problem, to combine the laws of Hydromechanics and Pneumatics so far as they will apply to the case in question.

In order that we may have as little confusion as possible in the symbols we may use in the development of formulæ we will group them together, at this time, as follows:—

D = the density, or weight, of a pound of steam at the indicated pressure.
d = the diameter of the pipe expressed in feet.
f = the *coefficient of friction;* a quantity to be determined by experiment.
g = the force of gravity exerted on all substance. (32.2.)
h = the head, or force by weight that induces the flow—expressed in feet.
L = the length of the pipe.
O = the area of the surface of the pipe. Frictional area.
p = the initial pressure (indicated boiler pressure).
p_1 = the final pressure, or pressure at point of delivery.
R = the total resistance of flow.
t = the temperature of the fluid.
v = the velocity of flow.
W = the volume of flow per second.
w = the weight of flow per second.
Q = the area of the cross-section of the pipe.
x = the frictional resistance estimated in pounds per sq. ft. of surface at a velocity of 1 ft. per second. A constant.

As early as 1648 Torricelli, in summing up the results of his many experiments on the flow of water through pipes, announced the theorem,—*the velocity flow of liquids is as the square root of the head.* This theorem modified by the influence of viscosity and friction, became the foundation of all Hydromechanics.

Expressed algebraically it would be $\frac{v^2}{2g} = h - z$ where (z) is the total drop in head at any particular moment of time. Omitting the value of (z) the equation becomes $v = \sqrt{2gh}$ (1). This is known in ordinary Dynamics as the ***theoretical velocity of discharge.***

Noting that the theoretical discharge was more than the actual Mariotti suggested that the difference between theory and the result of experiment was probably due to surface friction in the pipes through which it passed. Henri Pitot added the suggestion that the frictional resistance to flow was in inverse proportion to the

size of the pipe, and Dubant finally added viscosity to the resistance and promulgated the general theorem that *the loss in acceleration which water sustains when flowing through a pipe is equal to the resistance of friction or viscosity, or both.*

Daniel Bernoulli, from many experiments in his laboratory, set forth two theorems that, while they created much discussion and many debates among his relatives and contemporary scientists, really were the foundation upon which the theory of hydraulics is built.

The first theorem,—*the external and internal work done on a mass is equal to the change in kinetic energy produced.*

The second theorem,—the surface of a fluid which is discharging through an orifice at the bottom always remains horizontal. If the fluid mass is considered to be divided into an indefinite number of horizontal strata of the same bulk, and that these strata remain contiguous to each other, that all points descend perpendicularly with velocities inversely proportional to their depths.

The many field experiments of Coulomb, and the careful experiments he made under actual working conditions, enabled him to authoritatively set forth THE FIRST GENERAL LAW OF FLUID FRICTION. *The frictional resistance is independent of the pressure between the fluid and the solid against which it flows. But is directly proportional to the velocity, modified to meet the area and conditions of surface over which it flows.*

As viscosity has a very decided influence on the surface flow of water, and in steam it is nearly if not entirely absent, we can accept this law as directly applicable to saturated steam.

Expressing this law algebraically we would have $R = \dfrac{fwOv^2}{2g}$

THE COEFFICIENT OF FRICTION

Several experiments had been made by R. de Prony, Ettelwein, and others to determine the value of (f); and in case of water flowing at moderate speed through ordinary iron pipe they found f = .007567. So long as (f) depended upon velocity alone at moderate speed it gave quite accurate results.

Weisbach proposed the formula $4f = \dfrac{b}{\sqrt{v}} + a$ and gave the value (a) as .003598 and (b) as .004289.

H. P. G. Darcy, Inspector General of the Paris Water Works, carried out a great many experiments on the resistance of pipes to

the flow of water under varying conditions and with a degree of accuracy that makes his findings quite dependable. His coefficients were from the formula $f = a + \frac{b}{v}$ and gives to f the values of (a) and (b) as follows:

For 2″ pipe (f) = .00750	For 7″ pipe (f) = .00571
For 3″ pipe (f) = .00667	For 8″ pipe (f) = .00563
For 4″ pipe (f) = .00625	For 9″ pipe (f) = .00550
For 5″ pipe (f) = .00600	For 12″ pipe (f) = .00542
For 6″ pipe (f) = .00583	For 15″ pipe (f) = .00533

These results correspond very closely to the formula $f = .005 \left(1 + \frac{1}{12d}\right)$ which he suggests for general application. By experiment, however, he found that the results here given or the general formula was not exactly correct for widely differing velocities, and to meet all conditions he suggests $f = \left(a + \frac{d}{a_1}\right) + \frac{\left(b - \frac{b_1}{d^2}\right)}{v}$

in which the constants become

$a = .004346$
$a_1 = .0003992$
$b = .0010182$
$b_1 = .00005205$

It will be noticed from the above tables that (f) varies slightly with the diameter of the pipe, and recent demonstrations have shown that the power of velocity to which the resistance is proportioned is not exactly as the square, nor is it possible to determine the exact condition of the surface. It is not probable that any one can determine the exact carrying capacity of a steam pipe under all conditions; but we are limiting our investigations to water vapor, where the resistances are small compared to water itself and the velocities at the end differ only slightly from those at the beginning. Wrought iron or standard soft steel pipe has a moderately smooth surface, and as there is always some condensation present which adheres to the surface forming a film of water and offering the minimum resistance to the steam travel, the friction is less than that encountered by either air or water.

But as all resistances depend on velocity, diameter, and length of pipe, we will follow the development of the formulæ for flow

of water, and rely on the value of the coefficient of friction to express the difference in carrying qualities.

In the equation $v = \sqrt{2gh}$ or $h = \dfrac{v^2}{2g}$ no account is taken of friction. If we take (c) as a coefficient, the value of which can be determined by experiment, then the equation would be better expressed as $v = c\sqrt{2gh}$. But (h) actually consists of two parts,
h_1, that part expended to produce velocity, and
h_{11}, that part used to overcome friction.

If we separate (c) into the same relative values, c_1 and c_{11}, designating (c_{11}) as the coefficient of resistance, then

$$h_{11} = c_{11}h,\ v_1 = \sqrt{2gh_1} = \sqrt{\frac{2gh}{1 + c_{11}}}$$

$$c = \sqrt{\frac{1}{1 + c_1}} \qquad c_1 = \frac{1}{c_{11}} - 1$$

By actual experiment with the flow of water at ordinary velocity the value of c is about .97. Then c_1 would be .0628, which indicates that about 6 per cent. of the velocity head is taken up by friction.

Weisbach suggests as a general formula for the loss of energy

$E_f = \dfrac{f\,2v^2wL}{gd}$ which as you will note is based upon the laws of fluid friction as heretofore expressed. This has been accepted by all the noted authorities, but most of them have expressed the value of (f) in different terms.

Prof. Unwin, from many experiments, and deductions arrived at from the investigations of others, expresses the value as $f = K\left(1 + \dfrac{3}{10d}\right)$ where (K) is a constant whose value must be determined by experiment. If you will take the values of (f) as given by Darcy and bring them into the same form they would give $f = .005\left(1 + \dfrac{3}{12d}\right)$. Replacing .005 by (K), we will for the present use the formula $f=K\left(1 + \dfrac{3}{12d}\right)$ and substitute this

value in Weisbach's equation. As (p) represents the initial or boiler pressure, and (p_1) represents the pressure at C, or the final pressure, (p-p_1) must represent the head due to drop in pressure between terminals. As in practice the pressure is indicated per square inch, and as we have expressed (h) in feet, then 144 (p-p_1) $= hD$ or $p - p_1 = \frac{hD}{144}$. Where (w) represents the weight of steam flowing per second, and (h) represents the height in feet, then $E = wh = \frac{f\,2v^2wL}{gd}$. We will eliminate (w) from both sides of the equation and have $(h) = \frac{f\,2v^2L}{gd}$. Substituting $(p - p_1) = \frac{hD}{144}$ for the value of (h) we have $p - p_1 = \frac{f\,v^2DL}{72gd}$.

Placing $K\left(1 + \frac{3}{12d}\right)$ as the value of (f) we have $p - p_1 = \frac{Dv^2L}{72gd} \times K\left(1 + \frac{3}{12d}\right)$.

As we have considered (d), in all previous calculations, in terms of feet, and (w) in terms of seconds, we will now replace them with the same letters but having (d) represent inches, and (w) minutes. To do this we must give (d) a value of 1/12 the previous assignment, and (w) sixty times its previous value. Since (Q) represented the area in square feet, its value can be expressed by ($\frac{1}{4}\pi d^2$) or under the new value of (d) as $\frac{1}{4}\pi\left(\frac{d}{12}\right)^2 = \frac{\pi d^2}{576}$. Then as $w = QDv$ under the new values we should represent the equation by

$$60w = \frac{\pi d^2\,Dv}{576} \text{ or } w = \frac{\pi d^2\,Dv}{9.6}.$$ Then (v) would equal $\frac{9.6w}{\pi d^2D}$.

Substituting these values in the general equation for the value of

$(p\text{-}p_1)$ we would have $p\text{-}p_1 = \dfrac{DL\left(\dfrac{9.6w}{\pi d^2 D}\right)^2}{6gd} \times K\left(1 + \dfrac{3}{d}\right) =$

$$\frac{(9.6w)^2 L}{6\pi^2 d^5 gD} \times K\left(1 + \frac{3}{d}\right). \quad \text{or } w = \frac{\pi d^2}{9.6}\sqrt{\frac{6D\,(p\text{-}p_1)\,gd}{K\left(1 + \frac{3}{d}\right)L}}$$

THE GENERAL EQUATION

In our former equation $v = c\sqrt{2gh}$ we found that c_1 had a value of .0628 and the investigations of Darcy, Arson, Stockalpar, and others shows that this increases slightly in inverse proportion to the size of the pipe.

It is evident, therefore, that we can assume that (K) is a constant for all conditions, and that the variations of flow due to friction can be differentiated in the expression $\left(1 + \dfrac{3}{d}\right)$. Assuming, then, that (k) be of the value of .0028 our general formula would be better expressed as $w = 84d^2\sqrt{\dfrac{D\,(p\text{-}p_1)\,d}{\left(1 + \frac{3}{d}\right)L}}$ when the value of (g) is taken as 32.2 lbs.

Babcock and Unwin in their investigations assume the value of (k) as .0027 and modify it by $\left(1 + \dfrac{3.6}{d}\right)L$. For a 12 inch pipe this formula seems to agree very closely with the formula above. But it is evident that (K) is not a constant for all sizes of pipe. The experiments of all hydraulicians show plainly that there is a considerable variation, and that this variation is in inverse proportion to the size of pipe. If any degree of accuracy is to be obtained it must be taken into consideration.

Darcy, in his experiments in the flow of water, found this variation to be about 8 per cent. between a 12 inch pipe and a 2 inch pipe.

In making experiments with air flowing in pipes of different lengths Arson found the coefficient of friction to vary with the *velocity* and *diameter* of the pipe. This would change our formula

to $f = \frac{a}{v} + b$ and in this form he gives the values (a) and (b) as follows:

For a 2" pipe (a) = .04518, (b) = .01167
For a 3" pipe (a) = .03790, (b) = .00959
For a 4" pipe (a) = .03604, (b) = .00941
For a 10" pipe (a) = .01525, (b) = .00719
For a 12" pipe (a) = .00972, (b) = .00640

These values would make (f) for a 2 inch pipe .01212 and for a 12 inch pipe, .00650. By placing these values in a formula similar to the expression we have used it would approximate $f = .0059 (1 - \frac{3}{10d})$.

This shows that the variation of flow as effected by pipe sizes are about the same as that shown by Darcy except that the last term, which is always small except in small pipe, is considerably larger. Prof. Unwin, in commenting on the experiments of Cully and Sabine in carrying light packages through air tubes where great care is used to avoid friction, found that for lead pipes of 2¼ inch diameter and lengths varying from 2,000 to 6,000 feet that the value of (f) was approximately .007.

E. Stockalper, at the St. Gothard Tunnel, made many experiments with pipes of large diameter, and the results of his tests make the value of (f) as expressed by the Unwin formula .0028. This is considerably less than the results of Arson's experiments, and plainly shows that, for air, the size of the pipe very materially effects the friction. But we can not accept the theory that the friction of steam follows the action of air, except that it be superheated.

Joule established the law that if air expands without doing any external work *its temperature remains constant.* This is directly the opposite of the action of steam, for under no circumstances can steam expand without diminishing its temperature.

The more recent experiments plainly demonstrate that the frictional resistance of saturated steam is very much less than that of air, but that the size of the pipe has an influence on the velocity.

The kinetic energy necessary to move the steam at an increased velocity as it loses its temperature has, however, a very great effect if the increase in volume is considerable.

In any pipe of uniform size discharging steam against a back pressure the velocity of discharge is that due to the full drop in pres-

sure between (p) and (C) so long as the pressure at (C) is not less than .557 times the pressure at (p); and the amount of the discharge is, in that case the product of the velocity in the area of (d) and the density of the steam when expanded to the pressure at (C).

But when the pressure at (C) becomes 57.7 per cent. of that at (p) no further increase of discharge takes place with a further decrease in pressure at (C). For a full discussion of this phase see "The development of the DeLaval Turbine Engine," by Sir Alfred Ewing, KCB, LLD, FRS, Instructor Naval Engineering in Kings College, Cambridge.

In making up our tables and in determining the coefficient of friction we may assume that the variation of velocity at different points of any cross-section can be neglected. The steam may be considered as moving in uniform layers, (Bernoulli's Theorem) which are driven through the pipe against the frictional resistance by the difference of pressure at the terminals of the pipe.

Therefore we will consider the pressure as *constant*: each cross-section of pipe as of the same area, and the velocity to be the same at all points. Hence the velocity will be uniform and the frictional resistance a *constant*. This, as previously explained is not strictly true, but the acceleration of velocity is evidently not very great from the small differences of temperature at the terminals in any drop which we will consider; and in estimating the velocity of the steam at (C) to be the same as at (p) the error from not taking into consideration the extra friction caused by acceleration, will be nearly offset by the error of estimating the temperature of the pipe at the point of initial flow.

The recent developments of the turbine engine have called forth the most careful investigations of the influence of friction and its relations to velocity. While these results have been obtained from the flow through pipes of restricted lengths, we can see no reason why they should not hold good if the pipes were considerably extended.

We shall therefore accept these findings in the forming of our tables. In comparing the results of the most recent experiments of Ewing, with tests at Schenectady and Pittsburg, the value of (K) in our expression $K\left(1 + \frac{3}{d}\right)$ can be accepted as .0028 and treated as a constant in all sizes of pipe, and the variations of flow due to size of pipe and the kinetic energy used in acceleration can

best be expressed in modifications of the expression $\left(\frac{3}{d}\right)$.

Let (c_1), a variable quantity, replace the fixed quantity (3) in the above expression.

The equation reduced to its simplest form would then be $w = 84d^3 \sqrt{\frac{D\ (p-p_1)}{L\ (d + c_1)}}$. *The formula upon which the tables are based.* If, as in our previous equation, $c_1 = \left(\frac{1}{c_{11}} - 1\right)$ where (c_{11}) represents the variable coefficient of friction, we can, by giving to (c_{11}) the variations of value determined by experiments, readily find the value of (c_1).

Darcy shows in his experiments that the value of (c_{11}) varies 8 points in pipes ranging from 2 inches to 12 inches.

Arson, in experiments with air, shows practically the same variation. This is also confirmed by Stockalper. Ewing gives 10 points between 1 inch and 12 inches, and this variation seems to be confirmed by all the experiments that have come under the observation of the engineers with whom we have been able to confer.

Accepting this ratio as in inverse proportion to the size of the pipe the value of (c_{11}) becomes for a 1 inch pipe .2173 or 10/46. This renders the value of (c_1) as 3.6. Giving to (c_{11}) the values of 10/46 to 10/36 arranged in an arithmetical progression we have for the values of (c_1) the following:—

1 " pipe 3.6	3 " pipe 3.3	7" pipe 3.025
1¼" pipe 3.5	3½" pipe 3.25	8" pipe 2.95
1½" pipe 3.45	4 " pipe 3.2	9" pipe 2.825
2 " pipe 3.4	5 " pipe 3.15	10" pipe 2.75
2½" pipe 3.36	6 " pipe 3.1	12" pipe 2.6

These results are practically the same as those obtained by Unwin and Babcock had they made due allowance for the variation of flow, due to size of pipe, which is not a negligible quantity.

From such data as we have been able to gather this formula will not apply to superheated steam, as the friction seems to increase as temperature is added until with 100 degrees of superheat steam takes the properties of a gas and the frictional resistance must be based upon that for the flow of air.

In preparing the following tables for practical use it has been necessary to assume some conditions as constant, and consequently

to be taken into consideration ab initio. The frictional loss due to the steam entering the pipe from the nozzle of the boiler or another tee, being always present, has been estimated and need not receive further consideration; and the formula itself contemplates the horizontal position of the pipe. The loss of heat from exposure is a very considerable quantity, and, as economy of operation suggests a good quality, we have supposed the pipe to be covered with Standard Nonpareil High Pressure Covering, basing this loss upon (7.92) B.t.u. per sq. ft. per degree difference in temperature in 24 hours between the pipe and the surrounding temperature; which is supposed to be 70 degrees Fahrenheit. If other covers are used due allowance can be easily made. The frictional loss due to the placing of elbows, tees and valves in the line is to some extent an uncertain quantity. The frictional loss due to steam passing a reducing tee is given in the heading of each table, and is based upon the hydraulic equation for resistance for a simple orifice. It would probably be safe practice to estimate the resistance of a 45 degree elbow as 50 per cent. of the resistance of a tee; a 90 degree elbow as 70 per cent.; a Gate Valve as 100 per cent., and a Globe Valve as 160 per cent. In the formula ($p-p_1$) represents the hydraulic head due to drop in pressure between the terminals of the pipe, (w) represents the weight in pounds flowing per minute, D the density or weight of one cubic foot at the indicated pressure, and (c_1) a constant for each size of pipe, but varying inversely as the diameter. This quantity is given in the caption of each table. There seems to be some variation in the value of c_1 as the results of many experiments do not quite agree, and it is probable that the value we have assumed is not absolutely correct, but having analyzed the results of the many experiments in this country and compared them with those obtained in England, Germany and France, we can safely assert that for the size of pipe we have considered the difference will be so slight as to be negligible for ordinary calculations. The tables show the flow of saturated steam through pipes of various sizes and lengths and the velocity at different drops in pressure:

THE CARRYING CAPACITY OF STEAM PIPES

THE FLOW OF STEAM IN POUNDS PER MINUTE.

TABLE NO 1

INTERNAL DIA. .824" — C_1= 3.7 — 3/4" STANDARD PIPE. — L_1=1.5' — EXTERNAL DIA. 1.05"

L	B.P. 2.3 LBS D=.04352					B.P. 5.3 LBS D=.0507					B.P. 10.3 LBS D=.06253					B.P. 100.3 LBS D=.2617					B.P. 125.3 LBS D=.3147				
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1-OZ	2-OZ	4-OZ	8-OZ	16-OZ	1-OZ	2-OZ	4-OZ	8-OZ	16-OZ	1-OZ	2-OZ	4-OZ	8-OZ	16-OZ	1-OZ	2-OZ	4-OZ	8-OZ	16-OZ	1-OZ	2-OZ	4-OZ	8-OZ	16-OZ
1	.72	1.03	1.45	2.06	2.91	.78	1.11	1.57	2.22	3.14	.83	1.17	1.78	2.35	3.47	1.78	2.53	3.57	5.06	7.15	1.96	2.77	3.90	5.54	7.84
2	.61	.87	1.23	1.74	2.46	.66	.93	1.33	1.88	2.66	.73	1.03	1.46	2.07	2.93	1.51	2.13	3.02	4.27	6.04	1.65	2.34	3.31	4.69	6.63
3	.54	.76	1.08	1.53	2.17	.58	.83	1.17	1.66	2.34	.64	.91	1.29	1.83	2.58	1.33	1.88	2.66	3.76	5.33	1.46	2.06	2.92	4.13	5.84
5	.45	.63	.90	1.27	1.80	.48	.69	.97	1.38	1.95	.53	.76	1.07	1.52	2.15	1.10	1.56	2.21	3.13	4.43	1.21	1.72	2.43	3.44	4.86
7	.39	.55	.79	1.11	1.58	.42	.60	.85	1.20	1.70	.47	.66	.94	1.33	1.88	.97	1.37	1.94	2.74	3.88	1.05	1.49	2.11	2.99	4.24
10	.34	.48	.68	.96	1.36	.36	.51	.73	1.03	1.46	.41	.57	.83	1.14	1.66	.82	1.16	1.65	2.34	3.32	.90	1.28	1.81	2.57	3.64
15	.27	.39	.55	.79	1.12	.29	.41	.60	.84	1.22	.37	.46	.66	.94	1.34	.68	.97	1.38	1.95	2.77	.75	1.07	1.51	2.15	3.04
20	.25	.34	.48	.68	.98	.25	.37	.52	.75	1.06	.28	.40	.58	.82	1.17	.58	.84	1.19	1.70	2.41	.64	.92	1.31	1.87	2.65
25	.21	.30	.43	.62	.88	.23	.33	.47	.67	.95	.24	.33	.51	.75	1.04	.52	.75	1.07	1.53	2.17	.57	.82	1.17	1.67	2.38
30		.27	.39	.56	.80	.21	.30	.43	.61	.87	.22	.32	.46	.67	.95	.47	.68	.97	1.39	1.98	.52	.75	1.07	1.53	2.18
35		.25	.36	.52	.74		.27	.39	.56	.80	.20	.30	.43	.62	.88	.43	.63	.90	1.29	1.84	.47	.68	.98	1.41	2.05
40		.23	.33	.48	.69		.25	.36	.52	.75		.28	.40	.58	.83	.39	.58	.83	1.20	1.71	.44	.64	.92	1.32	1.88
45		.22	.31	.45	.65		.23	.34	.49	.71		.26	.38	.55	.78	.37	.54	.78	1.13	1.61	.40	.59	.86	1.23	1.77
50			.29	.42	.61		.22	.32	.47	.67		.24	.35	.51	.73	.34	.50	.73	1.06	1.52	.38	.56	.81	1.17	1.67
55			.27	.40	.58		.20	.30	.44	.63		.22	.33	.48	.70	.32	.48	.70	1.01	1.45	.35	.52	.76	1.11	1.59
60			.26	.38	.55			.28	.42	.60		.21	.32	.46	.67	.30	.45	.66	.96	1.38	.33	.50	.73	1.06	1.52
65			.25	.37	.53			.27	.40	.58			.29	.43	.63	.28	.43	.63	.92	1.32	.31	.45	.69	1.00	1.45
70				.34	.50			.25	.37	.55			.28	.42	.61	.26	.40	.59	.87	1.26	.29	.43	.66	.96	1.39
75				.33	.48				.36	.53			.27	.40	.58	.25	.38	.57	.84	1.22	.27	.42	.62	.92	1.33
80				.32	.47				.35	.51			.26	.39	.56		.37	.55	.81	1.18	.26	.40	.60	.89	1.29
85				.31	.45				.34	.49				.36	.54		.35	.53	.78	1.13		.38	.57	.85	1.24
90				.30	.44				.32	.47				.35	.52		.33	.51	.75	1.10		.36	.55	.82	1.20
95				.28	.42				.31	.45				.33	.50		.31	.48	.73	1.06		.34	.53	.79	1.16
100				.27	.40				.30	.44				.32	.49		.30	.47	.70	1.03		.33	.51	.77	1.13
120					.36				.28	.39				.28	.43		.25	.40	.61	.91		.27	.44	.67	1.00
140					.31					.34				.26	.41			.34	.54	.82			.39	.60	.91
160					.28					.30					.34			.29	.47	.74			.32	.53	.81
180										.28					.28			.25	.42	.67			.28	.47	.74
200																			.37	.60				.41	.65
225																			.32	.54				.36	.60
250																			.28	.49				.30	.52

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE

TABLE NO 2

INTERNAL DIA. 1.047" — $C_1 = 3.6$ — 1" STANDARD PIPE. — $L_1 = 3'$ — EXTERNAL DIA. 1.31"

L	B.P. 2.3 LBS. D = .04352					B.P. 5.3 LBS. D = .0507					B.P. 10.3 LBS. D = .06253				
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.
1	1.16	1.65	2.33	3.30	4.66	1.26	1.78	2.51	3.56	5.03	1.39	1.97	2.79	3.95	5.59
2	1.04	1.47	2.08	2.95	4.17	1.12	1.59	2.25	3.18	4.51	1.25	1.76	2.50	3.53	5.00
3	.95	1.35	1.90	2.69	3.81	1.00	1.45	2.05	2.91	4.11	1.14	1.61	2.28	3.22	4.57
5	.82	1.16	1.65	2.33	3.30	.89	1.26	1.78	2.52	3.55	.98	1.39	1.97	2.79	3.95
7	.73	1.04	1.47	2.08	2.95	.79	1.12	1.59	2.25	3.18	.88	1.25	1.76	2.50	3.53
10	.64	.91	1.29	1.83	2.59	.69	.98	1.39	1.97	2.79	.77	1.09	1.55	2.19	3.10
15	.55	.77	1.10	1.55	2.20	.59	.84	1.18	1.68	2.37	.64	.92	1.30	1.85	2.62
20	.47	.67	.96	1.36	1.94	.51	.73	1.04	1.47	2.09	.57	.81	1.15	1.63	2.32
25	.43	.61	.87	1.23	1.75	.46	.66	.94	1.33	1.89	.51	.73	1.04	1.48	2.10
30	.39	.56	.80	1.13	1.61	.42	.62	.86	1.23	1.74	.46	.66	.95	1.35	1.92
35	.36	.52	.74	1.06	1.50	.38	.55	.79	1.13	1.61	.43	.62	.88	1.26	1.79
40	.33	.48	.69	.99	1.40	.36	.52	.74	1.06	1.51	.40	.58	.83	1.18	1.68
45	.31	.45	.65	.93	1.33	.34	.49	.70	1.01	1.43	.38	.55	.78	1.12	1.59
50	.28	.42	.61	.87	1.25	.31	.46	.66	.95	1.35	.35	.51	.73	1.05	1.50
55	.27	.40	.58	.83	1.19	.30	.43	.63	.90	1.29	.33	.48	.70	1.00	1.44
60	.26	.38	.55	.80	1.14	.28	.41	.60	.86	1.24	.31	.46	.67	.96	1.38
65	.25	.37	.53	.77	1.10	.27	.40	.58	.83	1.19	.28	.45	.65	.93	1.32
70		.34	.50	.73	1.05	.25	.37	.55	.78	1.14	.27	.42	.61	.88	1.27
75		.33	.48	.70	1.01		.36	.53	.74	1.10	.26	.40	.59	.85	1.22
80		.32	.47	.68	.98		.35	.51	.72	1.06	.25	.38	.56	.81	1.17
85		.31	.45	.66	.95		.34	.49	.69	1.03		.37	.54	.79	1.14
90		.29	.43	.63	.91		.32	.47	.67	.99		.36	.53	.77	1.11
95		.28	.42	.61	.89		.31	.45	.65	.96		.35	.51	.75	1.08
100		.27	.41	.60	.87		.30	.44	.62	.94		.33	.49	.72	1.04
120			.36	.53	.78		.26	.39	.56	.84		.28	.43	.64	.93
140			.32	.48	.71			.35	.52	.77		.25	.38	.58	.85
160			.28	.43	.65			.31	.47	.71			.35	.54	.78
180			.26	.41	.61			.28	.43	.65			.30	.47	.77
200				.36	.56			.25	.40	.60			.27	.43	.66

L	B.P. 100.3 LBS. D .2617					B.P. 125.3 LBS. D = .3147				
	DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.
1	2.86	4.04	5.72	8.08	11.4	3.18	4.48	6.26	8.86	12.5
2	2.55	3.61	5.11	7.23	10.2	2.80	3.96	5.60	7.92	11.2
3	2.33	3.30	4.67	6.60	9.34	2.56	3.62	5.12	7.24	10.2
5	2.02	2.86	4.04	5.72	8.08	2.21	3.14	4.43	6.27	8.86
7	1.80	2.80	3.60	5.60	7.22	1.97	2.79	3.95	5.60	7.91
10	1.57	2.23	3.16	4.47	6.33	1.73	2.41	3.46	4.91	6.94
15	1.33	1.89	2.68	3.80	5.33	1.46	2.08	2.94	4.16	5.89
20	1.17	1.66	2.36	3.35	4.75	1.28	1.84	2.60	3.68	5.21
25	1.06	1.50	2.14	3.03	4.30	1.16	1.65	2.33	3.33	4.72
30	.96	1.37	1.96	2.78	3.95	1.06	1.52	2.16	3.06	4.34
35	.89	1.28	1.82	2.59	3.62	.98	1.41	2.00	2.84	4.11
40	.83	1.19	1.70	2.42	3.45		1.31	1.92	2.66	3.78
45	.78	1.12	1.61	2.29	3.26		1.24	1.77	2.52	3.60
50	.73	1.06	1.52	2.17	3.09		1.17	1.67	2.38	3.39
55	.70	1.01	1.45	2.07	2.95		1.11	1.59	2.27	3.24
60	.66	.95	1.38	1.97	2.82		1.06	1.52	2.17	3.10
65	.63	.92	1.32	1.90	2.71		1.00	1.45	2.08	2.93
70	.59	.87	1.26	1.81	2.60		.96	1.39	1.99	2.85
75	.57	.84	1.22	1.76	2.52			1.35	1.92	2.76
80	.54	.80	1.17	1.69	2.43			1.29	1.86	2.68
85	.53	.78	1.14	1.64	2.36			1.26	1.80	2.58
90	.51	.75	1.10	1.58	2.28			1.21	1.75	2.51
95	.48	.72	1.06	1.54	2.22			1.16	1.69	2.43
100	.46	.69	1.02	1.49	2.15			1.13	1.65	2.37
120	.39	.61	.91	1.34	1.94			1.00	1.55	2.14
140	.33	.53	.81	1.21	1.77				1.48	1.96
160	.28	.47	.73	1.10	1.68				1.34	1.80
180		.41	.66	1.01	1.51				1.15	1.67
200		.36	.60	.93	1.40				1.03	1.53

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE.

TABLE NO 3

INTERNAL DIA. 1.38 — C = 3.5 — $1\frac{1}{4}''$ STANDARD PIPE. — L = 4 — EXTERNAL DIA. 1.66

L	B.P. 2.3 LBS D = .04352					B.P. 5.3 LBS D = .0507					B.P. 10.3 LBS D = .06253					B.P. 100.3 LBS D = .2617					B.P. 125.3 LBS D = .3147				
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
1	2.33	3.30	4.66	6.59	9.31	2.51	3.55	5.05	7.11	10.3	2.79	3.94	5.58	7.89	11.1	5.71	8.08	11.4	16.1	22.8	6.26	8.86	12.5	17.7	25.0
2	2.12	3.00	4.25	6.01	8.50	2.29	3.49	4.59	6.98	9.18	2.55	3.60	5.10	7.21	10.2	5.21	7.37	10.4	14.7	20.8	5.71	8.10	11.4	16.2	22.8
3	1.97	2.78	3.94	5.56	7.87	2.12	3.00	4.26	6.01	8.50	2.35	3.24	4.71	6.68	9.42	4.83	6.83	9.66	13.6	19.3	5.29	7.49	10.6	15.0	21.1
5	1.73	2.45	3.47	4.91	6.94	1.87	2.64	3.74	5.29	7.49	2.08	2.94	4.16	5.58	8.33	4.26	6.02	8.52	12.0	17.0	4.67	6.61	9.34	13.2	18.6
7	1.57	2.22	3.14	4.44	6.28	1.69	2.39	3.39	4.79	6.78	1.88	2.66	3.77	5.32	7.54	3.85	5.45	7.70	10.9	15.4	4.21	5.97	8.43	11.9	16.8
10	1.39	1.96	2.78	3.93	5.57	1.50	2.12	3.00	4.24	6.01	1.67	2.36	3.34	4.71	6.68	3.39	4.82	6.80	9.65	13.6	3.78	5.29	7.48	10.6	14.9
15	1.19	1.69	2.30	3.38	4.78	1.28	1.81	2.64	3.68	5.15	1.43	2.02	2.86	4.05	5.72	2.92	4.14	5.85	8.29	11.7	3.20	4.54	6.41	9.09	12.8
20	1.05	1.49	2.11	3.00	4.24	1.18	1.61	2.28	3.23	4.58	1.25	1.78	2.58	3.58	5.08	2.58	3.67	5.29	7.36	10.4	2.88	4.08	5.69	8.08	11.4
25	.96	1.35	1.92	2.72	3.86	1.03	1.46	2.07	2.94	4.16	1.14	1.62	2.30	3.26	4.62	2.35	3.33	4.78	6.66	9.48	2.60	3.66	5.23	7.34	10.4
30	.89	1.26	1.78	2.51	3.56	.94	1.34	1.98	2.70	3.84	1.05	1.49	2.12	3.01	4.26	2.16	3.07	4.35	6.27	8.73	2.38	3.38	4.79	6.78	9.59
35	.82	1.17	1.66	2.35	3.33	.88	1.25	1.78	2.52	3.58	.98	1.39	1.98	2.80	3.98	2.01	2.86	4.06	5.75	8.15	2.21	3.18	4.45	6.30	8.94
40	.76	1.09	1.55	2.20	3.12	.82	1.17	1.67	2.37	3.37	.92	1.31	1.86	2.64	3.75	1.88	2.73	3.85	5.41	7.60	2.07	2.95	4.18	5.94	8.41
45	.72	1.05	1.46	2.08	2.95	.77	1.10	1.57	2.24	3.18	.87	1.24	1.76	2.50	3.55	1.78	2.54	3.61	5.12	7.26	1.95	2.78	3.96	5.61	7.96
50	.68	.95	1.38	2.00	2.80	.73	1.05	1.50	2.18	3.08	.82	1.17	1.67	2.37	3.37	1.69	2.41	3.43	4.87	6.92	1.85	2.65	3.76	5.36	7.56
55	.64	.92	1.31	1.89	2.62	.70	1.00	1.48	2.04	2.90	.78	1.12	1.59	2.27	3.22	1.60	2.29	3.26	4.64	6.59	1.76	2.51	3.59	5.09	7.24
60	.62	.89	1.27	1.81	2.57	.66	.95	1.36	1.94	2.77	.74	1.06	1.52	2.14	3.08	1.52	2.19	3.12	4.45	6.31	1.68	2.40	3.43	4.88	6.98
65	.59	.85	1.22	1.75	2.47	.63	.91	1.31	1.87	2.66	.70	1.02	1.46	2.05	2.96	1.45	2.10	3.00	4.28	6.08	1.61	2.31	3.30	4.69	6.68
70	.56	.81	1.17	1.67	2.36	.61	.88	1.26	1.81	2.57	.67	.97	1.40	1.99	2.84	1.40	2.02	2.89	4.12	5.86	1.54	2.23	3.17	4.52	6.43
75	.54	.79	1.13	1.62	2.30	.59	.85	1.22	1.75	2.49	.65	.94	1.35	1.93	2.76	1.35	1.95	2.79	3.99	5.66	1.49	2.15	3.07	4.38	6.22
80	.52	.76	1.09	1.56	2.23	.56	.81	1.17	1.68	2.40	.64	.92	1.32	1.88	2.67	1.30	1.89	2.69	3.85	5.48	1.43	2.07	2.96	4.23	6.02
85	.50	.73	1.05	1.51	2.15	.54	.79	1.14	1.63	2.33	.61	.88	1.27	1.82	2.60	1.26	1.82	2.61	3.74	5.32	1.39	2.01	2.88	4.11	5.85
90	.48	.71	1.02	1.47	2.10	.52	.76	1.10	1.58	2.26	.58	.85	1.23	1.76	2.52	1.21	1.76	2.53	3.63	5.17	1.33	1.93	2.78	3.97	5.67
95	.47	.69	.99	1.43	2.04	.50	.73	1.07	1.53	2.20	.56	.82	1.19	1.71	2.45	1.18	1.71	2.46	3.53	5.08	1.29	1.88	2.70	3.87	5.52
100	.45	.67	.97	1.39	1.99	.49	.71	1.04	1.49	2.14	.55	.80	1.16	1.67	2.39	1.14	1.66	2.39	3.46	4.90	1.25	1.82	2.62	3.76	5.37
120	.39	.59	.86	1.25	1.80	.43	.64	.93	1.35	1.94	.48	.71	1.04	1.50	2.15	1.01	1.49	2.16	3.11	4.45	1.11	1.64	2.37	3.42	4.89
140	.35	.53	.78	1.14	1.65	.38	.58	.85	1.24	1.79	.43	.64	.90	1.38	1.99	.91	1.35	1.98	2.89	4.11	.99	1.48	2.16	3.13	4.49
160	.32	.48	.72	1.06	1.53	.33	.52	.77	1.14	1.65	.38	.58	.87	1.27	1.85	.81	1.23	1.81	2.64	3.80	.90	1.35	1.99	2.90	4.18
180	.25	.44	.66	.98	1.43	.30	.47	.71	1.06	1.54	.34	.53	.80	1.18	1.72	.74	1.18	1.68	2.46	3.56	.83	1.25	1.86	2.71	3.95
200	.26	.40	.61	.92	1.34	.27	.42	.66	.98	1.45	.30	.48	.74	1.10	1.62	.67	1.04	1.57	2.33	3.36	.74	1.14	1.72	2.53	3.63

THE CARRYING CAPCITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE.

TABLE NO 4

INTERNAL DIA. 1.61" C_1=3.45 1 1/2" STANDARD PIPE. L_1=5' EXTERNAL DIA. 1.9"

L	B.P. 2.3 LBS		D=.04352			B.P. 5.3 LBS		D=.0507			B.P. 10.3 LBS		D=.06253			B.P. 100.3 LBS		D=.2617			B.P. 125.3 LBS		D=.3147		
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
2	3.07	4.34	6.14	8.69	12.3	3.31	4.68	6.63	9.37	13.2	3.68	5.20	7.36	10.4	14.7	7.53	10.6	15.0	21.2	30.1	8.26	11.6	16.5	23.3	33.4
5	2.57	3.63	5.14	7.27	10.2	2.77	3.66	5.54	7.83	11.0	3.07	4.35	6.15	8.71	12.3	6.30	8.90	12.6	17.8	25.2	6.91	9.78	13.8	19.5	27.6
10	2.10	2.96	4.20	5.93	8.40	2.26	3.20	4.52	6.40	9.05	2.51	3.55	5.03	7.10	10.0	5.13	7.27	10.2	14.5	20.5	5.59	8.97	11.2	15.9	22.5
15	1.81	2.57	3.63	5.14	7.27	1.95	2.76	3.91	5.53	7.83	2.17	3.07	4.35	6.15	8.71	4.44	6.25	8.89	12.6	17.8	4.89	6.89	9.76	13.8	19.5
20	1.59	2.29	3.23	4.59	6.48	1.74	2.47	3.50	4.95	7.02	1.93	2.74	3.88	5.50	7.78	3.96	5.61	7.95	11.2	15.9	4.35	6.19	8.73	12.3	17.4
25	1.46	2.09	2.95	4.19	5.92	1.59	2.24	3.19	4.52	6.39	1.76	2.50	3.54	5.02	7.10	3.61	5.11	7.25	10.2	14.5	3.96	5.61	7.95	11.2	15.9
30	1.35	1.97	2.78	3.86	5.48	1.46	2.07	2.94	4.16	5.91	1.62	2.30	3.27	4.63	6.56	3.33	4.72	6.70	9.48	13.4	3.65	5.18	7.35	10.4	14.7
35	1.26	1.79	2.55	3.61	5.12	1.36	1.94	2.75	3.90	5.52	1.51	2.16	3.05	4.34	6.18	3.11	4.41	6.26	8.87	12.5	3.41	4.84	6.87	9.72	13.7
40	1.18	1.69	2.39	3.41	4.82	1.27	1.81	2.58	3.66	5.20	1.42	2.02	2.87	4.07	5.78	2.91	4.15	5.88	8.35	11.8	3.20	4.51	6.47	9.17	13.0
45	1.12	1.60	2.27	3.23	4.57	1.20	1.72	2.44	3.47	4.92	1.34	1.91	2.72	3.86	5.48	2.76	3.93	5.58	7.92	11.2	3.08	4.31	6.12	8.68	12.3
50	1.06	1.52	2.16	3.07	4.35	1.15	1.62	2.33	3.31	4.70	1.27	1.81	2.61	3.67	5.21	2.62	3.74	5.31	7.54	10.7	2.88	4.11	5.83	8.28	11.7
55	1.02	1.45	2.07	2.94	4.17	1.10	1.53	2.23	3.17	4.49	1.21	1.73	2.47	3.51	4.99	2.50	3.55	5.07	7.20	10.2	2.75	3.92	5.57	7.91	11.2
60	.96	1.38	1.97	2.81	3.99	1.04	1.49	2.13	3.03	4.31	1.16	1.66	2.37	3.37	4.79	2.39	3.41	4.86	6.91	9.80	2.63	3.75	5.34	7.58	10.7
65	.93	1.33	1.90	2.71	3.84	1.00	1.44	2.05	2.92	4.15	1.12	1.60	2.28	3.25	4.61	2.30	3.28	4.68	6.65	9.44	2.53	3.61	5.14	7.31	10.3
70	.89	1.28	1.83	2.61	3.71	.96	1.36	1.97	2.81	4.00	1.07	1.53	2.20	3.12	4.45	2.21	3.16	4.51	6.41	9.12	2.43	3.48	4.95	7.05	9.99
75	.86	1.23	1.77	2.52	3.59	.93	1.33	1.91	2.72	3.87	1.04	1.48	2.13	3.03	4.31	2.13	3.05	4.36	6.20	8.82	2.34	3.35	4.78	6.80	9.66
80	.83	1.19	1.71	2.46	3.48	.90	1.29	1.85	2.63	3.75	.99	1.43	2.05	2.92	4.16	2.06	2.95	4.22	6.01	8.55	2.27	3.25	4.64	6.60	9.37
85	.80	1.16	1.66	2.37	3.38	.87	1.25	1.79	2.56	3.64	.96	1.39	1.99	2.84	4.04	2.00	2.86	4.10	5.83	8.30	2.19	3.14	4.49	6.40	9.10
90	.77	1.12	1.60	2.30	3.27	.84	1.21	1.74	2.48	3.54	.93	1.34	1.93	2.75	3.93	1.93	2.78	3.97	5.57	8.06	2.13	3.01	4.37	6.23	8.85
95	.75	1.09	1.56	2.24	3.19	.81	1.17	1.69	2.41	3.45	.90	1.30	1.87	2.68	3.82	1.88	2.70	3.87	5.52	7.86	2.06	2.97	4.25	6.06	8.62
100	.73	1.05	1.52	2.17	3.17	.78	1.14	1.64	2.35	3.35	.87	1.26	1.82	2.60	3.72	1.81	2.62	3.76	5.37	7.65	2.00	2.88	4.13	5.89	8.39
120	.64	.95	1.37	1.96	2.83	.70	1.02	1.48	2.13	3.05	.78	1.14	1.66	2.37	3.39	1.63	2.37	3.41	4.89	6.98	1.80	2.66	3.75	5.37	7.66
140	.58	.86	1.26	1.82	2.60	.63	.94	1.36	1.97	2.82	.70	1.09	1.51	2.28	3.13	1.47	2.16	3.13	4.70	6.44	1.63	2.38	3.45	4.95	7.07
160	.53	.79	1.16	1.69	2.43	.57	.85	1.26	1.82	2.64	.63	.95	1.39	2.02	2.91	1.35	1.98	2.90	4.17	6.01	1.49	2.19	3.19	4.52	6.51
180	.47	.72	1.07	1.57	2.27	.52	.79	1.16	1.70	2.45	.58	.88	1.30	1.89	2.73	1.23	1.84	2.70	3.91	5.63	1.37	2.04	2.98	4.31	6.19
200	.43	.67	1.00	1.47	2.14	.47	.72	1.08	1.59	2.31	.52	.81	1.20	1.77	2.56	1.14	1.71	2.53	3.68	5.32	1.26	1.89	2.79	4.05	5.84
225	.39	.60	.92	1.36	1.95	.42	.66	1.00	1.48	2.16	.47	.73	1.11	1.64	2.39	1.03	1.57	2.35	3.43	4.98	1.15	1.74	2.59	3.78	5.47
250	.34	.56	.85	1.26	1.87	.37	.60	.92	1.38	2.09	.42	.67	1.08	1.53	2.25	.93	1.44	2.17	3.20	4.67	1.04	1.61	2.41	3.55	5.15
275	.30	.50	.79	1.19	1.76	.33	.55	.85	1.20	1.90	.38	.62	.96	1.44	2.12	.84	1.33	2.08	3.02	4.41	.94	1.48	2.25	3.33	4.86
300	.26	.46	.73	1.12	1.66	.29	.50	.79	1.21	1.70	.29	.51	.86	1.35	1.98	.76	1.23	1.80	2.84	4.18	.86	1.37	2.11	3.15	4.61

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE. TABLE NO 5

INTERNAL DIA. 2.067" $C_1 = 3.4$ 2" STANDARD PIPE. $L_1 = 7'$ EXTERNAL DIA. 2.375"

L	B.P. 2.3 LBS D=.04352					B.P. 5.3 LBS D=.0507					B.P. 10.3 LBS D=.06258					B.P. 100.3 LBS D=.2617					B.P. 125.3 LBS D=.3147				
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
2	5.51	7.78	11.0	15.5	22.7	5.95	8.41	11.9	16.8	23.8	6.60	10.5	13.2	21.2	26.4	13.5	21.6	27.0	43.8	54.0	16.8	23.8	33.6	47.6	62.2
5	4.77	6.75	9.55	13.5	19.1	5.15	7.29	10.3	14.9	20.6	5.72	8.08	11.4	16.1	22.8	11.7	16.5	23.4	33.1	46.8	12.8	18.1	25.6	36.2	51.3
10	4.01	5.67	8.02	11.3	16.0	4.29	6.11	8.60	12.2	17.3	4.79	6.79	9.60	13.5	19.2	9.88	13.9	19.6	27.8	39.3	10.7	15.2	21.5	30.5	43.1
15	3.52	4.98	7.05	9.97	14.1	3.79	5.37	7.60	10.7	15.2	4.22	5.96	8.45	11.9	16.9	8.63	12.2	17.2	24.4	34.5	9.46	13.4	18.9	26.8	37.9
20	3.17	4.49	6.35	8.99	12.7	3.42	4.85	6.86	9.72	13.7	3.79	5.37	7.61	10.8	15.2	7.78	11.0	15.5	22.0	31.2	8.53	12.0	17.0	24.1	34.2
25	2.92	4.12	5.83	8.25	11.6	3.14	4.44	6.30	8.90	12.6	3.48	4.93	6.99	9.89	14.0	7.13	10.1	14.3	20.2	28.6	7.85	11.0	15.6	22.2	31.3
30	2.69	3.82	5.41	7.66	10.8	2.93	4.12	5.84	8.29	11.7	3.22	4.57	6.48	9.19	12.9	6.63	9.39	13.3	18.8	26.6	7.26	10.3	14.5	20.6	29.2
35	2.53	3.59	5.09	7.20	10.2	2.73	3.87	5.49	7.76	11.0	3.02	4.29	6.08	8.62	12.2	6.20	8.80	12.1	17.6	24.9	6.83	9.64	13.7	19.3	27.4
40	2.38	3.38	4.79	6.79	9.64	2.57	3.65	5.18	7.33	10.3	2.86	4.05	5.75	8.14	11.5	5.86	8.31	11.7	16.6	23.6	6.42	9.11	12.9	18.2	25.9
45	2.26	3.24	4.55	6.45	9.14	2.44	3.47	4.92	6.97	9.88	2.70	3.85	5.45	7.74	10.9	5.56	7.89	11.1	15.8	22.4	6.09	8.65	12.2	17.3	24.6
50	2.15	3.05	4.34	6.15	8.72	2.32	3.30	4.69	6.64	9.42	2.58	3.67	5.21	7.38	10.4	5.37	7.53	10.7	15.1	21.4	5.80	8.25	11.7	16.5	23.4
55	2.06	2.92	4.16	5.89	8.36	2.22	3.16	4.49	6.37	9.02	2.47	3.51	4.99	7.07	10.0	5.07	7.20	10.2	14.5	20.5	5.56	7.90	11.2	15.9	22.5
60	1.98	2.81	4.00	5.67	8.04	2.13	3.03	4.31	6.11	8.67	2.37	3.37	4.79	6.80	9.63	4.86	6.92	9.82	13.9	19.7	5.33	7.58	10.7	15.2	21.6
65	1.91	2.71	3.86	5.46	7.76	2.05	2.92	4.16	5.90	8.37	2.28	3.25	4.62	6.55	9.29	4.69	6.57	9.48	13.4	19.0	5.13	7.30	10.3	14.7	20.8
70	1.83	2.61	3.72	5.27	7.49	1.98	2.82	4.01	5.70	8.08	2.19	3.18	4.45	6.32	8.97	4.52	6.44	9.15	12.9	18.4	4.95	7.05	10.0	14.3	20.1
75	1.80	2.53	3.65	5.11	7.35	1.91	2.72	3.93	5.51	7.82	2.13	3.08	4.32	6.18	8.70	4.36	6.23	8.48	12.7	17.8	4.79	6.83	9.64	13.8	19.5
80	1.71	2.44	3.48	4.95	7.02	1.85	2.64	3.76	5.35	7.59	2.05	2.93	4.18	5.94	8.43	4.22	6.08	8.37	12.1	17.2	4.63	6.61	9.40	13.3	18.9
85	1.66	2.37	3.38	4.87	6.88	1.79	2.56	3.65	5.19	7.37	1.99	2.85	4.06	5.77	8.19	4.11	5.86	8.24	11.8	16.8	4.51	6.43	9.15	13.0	18.4
90	1.60	2.30	3.28	4.67	6.64	1.74	2.49	3.55	5.05	7.17	1.94	2.77	3.95	5.62	7.97	3.99	5.70	8.11	11.5	16.3	4.37	6.25	8.89	12.6	17.9
95	1.56	2.24	3.20	4.55	6.47	1.68	2.42	3.45	4.92	6.98	1.90	2.69	3.89	5.47	7.77	3.88	5.55	7.90	11.2	15.9	4.24	6.07	8.64	12.3	17.4
100	1.52	2.19	3.12	4.45	6.32	1.64	2.35	3.37	4.79	6.82	1.82	2.61	3.74	5.32	7.57	3.77	5.40	7.69	10.9	15.5	4.13	5.92	8.43	12.0	17.0
120	1.37	1.98	2.84	4.05	5.77	1.49	2.14	3.07	4.38	6.24	1.65	2.38	3.41	4.87	6.93	3.42	4.91	7.03	10.0	14.2	3.75	5.39	7.70	10.6	15.6
140	1.26	1.82	2.62	3.75	5.35	1.36	1.97	2.83	4.05	5.77	1.51	2.19	3.14	4.50	6.41	3.13	4.52	6.48	9.26	13.1	3.45	4.97	7.12	10.3	14.4
160	1.15	1.69	2.48	3.50	4.99	1.25	1.82	2.63	3.77	5.39	1.39	2.04	2.92	4.28	5.99	2.90	4.20	6.04	8.62	12.3	3.19	4.62	6.63	9.49	13.5
180	1.07	1.57	2.28	3.28	4.70	1.16	1.65	2.47	3.45	5.08	1.28	1.89	2.78	3.94	5.63	2.69	3.92	5.66	8.12	11.6	2.95	4.32	6.22	8.93	12.7
200	.99	1.47	2.14	3.10	4.44	1.08	1.59	2.32	3.35	4.80	1.20	1.78	2.58	3.78	5.34	2.53	3.90	5.35	7.69	10.9	2.78	4.05	5.87	8.44	12.0
225	.91	1.36	1.90	2.89	4.17	.99	1.47	2.16	3.13	4.50	1.10	1.64	2.40	3.48	5.00	2.33	3.44	5.00	7.21	10.3	2.57	3.75	5.49	7.93	11.3
250	.84	1.26	1.87	2.72	3.98	.91	1.37	2.02	2.95	4.25	1.01	1.52	2.26	3.27	4.72	2.16	3.21	4.69	6.79	9.75	2.38	3.54	5.16	7.47	10.7
275	.77	1.18	1.75	2.57	3.72	.84	1.25	1.90	2.73	4.08	.95	1.44	2.18	3.11	4.49	2.00	3.02	4.42	6.48	9.25	2.22	3.32	4.87	7.07	10.1
300	.72	1.11	1.66	2.45	3.55	.77	1.20	1.79	2.64	3.88	.87	1.33	2.00	2.96	4.26	1.86	2.83	4.19	6.11	8.82	2.07	3.17	4.68	6.71	9.69

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE.

TABLE NO 6

INTERNAL DIA. 2.467″ $C_1 = 3.6$ 2½″ STANDARD PIPE. $L_1 = 10$ EXTERNAL DIA. 2.875″

L	B.P. 2.3 LBS				D = .04352	B.P. 5.3 LBS				D = .0507	B.P. 10.3 LBS				D = .06258	B.P. 100.3 LBS				D = .2617	B.P. 125.3 LBS				D = .3147
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
5	7.03	9.94	14.0	19.8	28.1	7.59	10.7	15.2	21.4	30.3	8.20	11.9	16.4	23.3	32.8	17.2	24.3	34.4	48.7	68.9	18.9	26.7	37.8	53.5	75.6
10	6.09	8.62	12.1	17.2	24.3	6.56	9.29	13.2	18.6	26.2	7.29	10.3	14.6	20.6	29.2	14.9	21.1	29.8	42.2	59.7	16.3	23.1	32.7	46.3	65.5
15	5.43	7.69	10.8	15.4	21.7	5.86	8.31	11.7	16.6	23.5	6.52	9.24	13.0	18.5	26.1	13.3	18.8	26.6	37.7	53.3	14.5	20.6	29.3	41.3	58.3
20	4.96	7.02	9.93	14.0	19.8	5.35	7.57	10.7	15.1	21.4	5.94	8.42	11.9	16.8	23.8	12.1	17.2	24.3	34.4	48.7	13.3	18.8	26.6	37.6	53.4
25	4.58	6.50	9.18	13.0	18.4	4.93	6.99	9.90	14.0	19.8	5.50	7.79	11.0	15.6	22.0	11.2	15.9	22.5	31.1	45.1	12.2	17.4	24.6	34.9	49.4
30	4.28	6.07	8.59	12.1	17.2	4.62	6.54	9.27	13.1	18.5	5.13	7.28	10.3	14.6	20.6	10.5	14.8	21.0	29.8	42.1	11.4	16.3	23.0	32.7	46.2
35	4.03	5.71	8.09	11.4	16.2	4.38	6.20	8.78	12.3	17.5	4.84	6.86	9.72	13.7	19.4	9.84	14.0	19.7	28.0	39.5	10.8	15.3	21.7	30.8	43.6
40	3.82	5.42	7.68	10.8	15.4	4.18	5.84	8.30	11.7	16.6	4.58	6.49	9.21	13.0	18.4	9.37	13.2	18.8	26.6	37.7	10.3	14.6	20.6	29.3	41.4
45	3.63	5.15	7.30	10.3	14.6	3.96	5.57	7.89	11.1	15.8	4.36	6.19	8.77	12.4	17.5	8.92	12.5	17.9	25.3	35.9	9.79	13.8	19.6	27.8	39.4
50	3.48	4.93	6.99	9.91	14.0	3.74	5.31	7.54	10.6	15.1	4.17	5.86	8.39	11.9	16.8	8.52	12.1	17.1	24.2	34.3	9.31	13.2	18.6	26.6	37.7
60	3.20	4.55	6.45	9.16	12.9	3.46	4.92	6.97	9.89	14.0	3.84	5.46	7.75	11.0	15.5	7.85	11.1	15.8	22.4	31.7	8.64	12.3	17.4	24.7	34.9
70	2.98	4.24	6.02	8.55	12.1	3.22	4.58	6.51	9.22	13.0	3.58	5.04	7.24	10.1	14.5	7.33	10.4	14.9	20.9	29.7	8.06	11.4	16.2	23.0	32.6
80	2.80	3.99	5.67	8.05	11.4	3.03	4.31	6.13	8.69	12.3	3.36	4.79	6.81	9.66	13.7	6.88	9.80	13.9	19.7	28.0	7.72	10.7	15.3	21.7	30.7
90	2.64	3.77	5.37	7.62	10.8	2.86	4.08	5.80	8.24	11.6	3.17	4.53	6.44	9.15	12.9	6.50	9.25	13.1	18.7	26.5	7.16	10.2	14.5	20.5	29.1
100	2.50	3.58	5.10	7.26	10.3	2.71	3.87	5.52	7.83	11.1	3.02	4.30	6.18	8.71	12.3	6.16	8.80	12.5	17.8	25.2	6.80	9.69	13.7	19.5	27.7
120	2.28	3.28	4.67	6.66	9.45	2.47	3.53	5.05	7.18	10.2	2.74	3.93	5.60	7.98	11.3	5.61	8.03	11.4	16.3	23.1	6.20	8.86	12.6	17.9	25.4
140	2.10	3.02	4.33	6.17	8.78	2.27	3.26	4.67	6.66	9.47	2.53	3.68	5.20	7.40	10.5	5.18	7.44	10.5	15.1	21.5	5.73	8.20	11.7	16.6	23.6
160	1.96	2.82	4.05	5.78	8.23	2.11	3.04	4.36	6.23	8.86	2.34	3.38	4.85	6.93	9.86	4.85	6.98	9.97	14.2	20.2	5.32	7.65	10.9	15.6	22.1
180	1.82	2.64	3.80	5.44	7.75	1.97	2.85	4.10	5.87	8.37	2.19	3.17	4.56	6.52	9.30	4.58	6.58	9.37	13.3	19.0	4.98	7.18	10.3	14.7	20.9
200	1.71	2.50	3.59	5.17	7.35	1.84	2.68	3.87	5.55	7.93	2.05	2.99	4.31	6.18	8.82	4.26	6.17	8.87	12.6	18.0	4.69	6.78	9.64	13.9	19.8
225	1.58	2.32	3.36	4.84	6.91	1.70	2.50	3.62	5.21	7.45	1.91	2.79	4.04	5.81	8.30	3.97	5.78	8.32	11.9	17.0	4.36	6.34	9.14	13.1	18.7
250	1.48	2.18	3.17	4.57	6.55	1.54	2.35	3.31	4.92	7.06	1.77	2.61	3.80	5.48	7.86	3.72	5.44	7.84	11.3	16.1	4.09	5.97	8.63	12.4	17.7
275	1.38	2.05	2.99	4.33	6.22	1.49	2.21	3.23	4.67	6.71	1.66	2.46	3.60	5.20	7.47	3.48	5.12	7.44	10.7	15.3	3.83	5.63	8.17	11.7	16.8
300	1.28	1.93	2.83	4.12	5.92	1.39	2.09	3.02	4.45	6.40	1.56	2.32	3.41	4.95	7.12	3.28	4.85	7.07	10.2	14.6	3.61	5.33	7.77	11.2	16.1
350	1.00	1.73	2.57	3.76	5.44	1.22	1.87	2.97	4.06	5.97	1.37	2.08	3.09	4.52	6.44	2.97	4.38	6.43	9.36	13.9	3.22	4.82	7.08	10.3	14.8
400	.89	1.56	2.34	3.46	5.02	1.08	1.78	2.58	3.78	5.43	1.21	1.88	2.82	4.16	6.05	2.59	3.95	5.89	8.61	12.5	2.88	4.38	6.50	9.49	13.7
450	.79	1.41	2.16	3.21	4.70	.96	1.52	2.33	3.46	5.07	1.07	1.60	2.59	3.86	5.64	2.34	3.63	5.45	8.08	11.6	2.59	4.01	6.00	8.84	12.8
500	.59	1.27	1.99	2.98	4.41	.84	1.38	2.14	3.22	4.74	.94	1.54	2.39	3.59	5.29	2.10	3.38	5.06	7.51	10.9	2.38	3.67	5.57	8.26	12.0
600	.42	1.05	1.69	2.61	3.90	.64	1.18	1.88	2.81	4.21	.72	1.27	2.04	3.14	4.69	1.67	2.84	4.38	6.61	9.79	1.86	3.09	4.88	7.28	10.7
700		.89	1.44	2.29	3.49	.44	.92	1.56	2.48	3.77	.52	1.03	1.75	2.77	4.20	1.28	1.84	3.88	5.89	8.82	1.46	2.60	4.21	6.49	9.71

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE. TABLE NO 7

INTERNAL DIA. 3.066 C_1= 3.3 3" STANDARD PIPE. L_1= 14' EXTERNAL DIA. 3.5

L	B.P. 2.3 LBS D=.04352					B.P. 5.3 LBS D=.0507					B.P. 10.3 LBS D=.06253				
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
10	10.2	14.4	20.4	28.9	40.8	11.0	15.5	22.0	31.1	44.0	12.2	17.3	24.4	34.6	48.9
20	8.56	12.1	17.1	24.2	34.3	9.24	12.9	18.4	26.1	37.0	10.1	14.4	20.4	29.0	41.1
30	7.51	10.6	15.0	21.3	30.1	8.11	11.4	16.2	22.9	32.5	9.01	12.7	18.0	25.5	36.1
40	6.77	9.59	13.6	19.2	27.2	7.31	10.3	14.6	20.7	29.3	8.12	11.5	16.2	23.0	32.6
50	6.20	8.80	12.4	17.6	25.0	6.70	9.50	13.4	19.0	26.9	7.44	10.5	14.9	21.1	29.9
60	5.76	8.17	11.5	16.4	23.2	6.22	8.81	12.5	17.7	25.0	6.90	9.79	13.8	19.6	27.8
70	5.39	7.65	10.8	15.3	21.7	5.82	8.26	11.7	16.6	23.5	6.46	9.17	13.0	18.4	26.1
80	5.07	7.22	10.3	14.5	20.5	5.49	7.80	11.0	15.6	22.2	6.09	8.66	12.2	17.4	24.6
90	4.81	6.84	9.72	13.7	19.5	5.20	7.39	10.5	14.8	21.1	5.77	8.21	11.6	16.5	23.4
100	4.58	6.52	9.27	13.1	18.6	4.94	7.04	10.0	14.2	20.1	5.49	7.82	11.1	15.7	22.3
120	4.20	5.99	8.52	12.1	17.1	4.53	6.47	9.19	13.0	18.5	5.04	7.18	10.2	14.5	20.5
140	3.89	5.56	7.92	11.2	15.9	4.20	6.01	8.55	12.1	17.2	4.67	6.67	9.50	13.5	19.1
160	3.68	5.20	7.42	10.5	15.0	3.92	5.62	8.01	11.4	16.2	4.36	6.25	8.91	12.6	18.0
180	3.41	4.89	7.00	9.97	14.1	3.67	5.30	7.56	10.8	15.3	4.09	5.88	8.40	11.9	17.0
200	3.22	4.63	6.64	9.47	13.5	3.48	5.00	7.17	10.2	14.4	3.86	5.57	7.96	11.3	16.1
225	3.01	4.35	6.25	8.98	12.7	3.25	4.70	6.75	9.64	13.7	3.62	5.23	7.50	10.7	15.2
250	2.88	4.00	5.91	8.46	12.0	3.05	4.48	6.38	9.14	13.0	3.40	4.98	7.09	10.1	14.4
275	2.67	3.80	5.61	8.05	11.5	2.87	4.19	6.05	8.68	12.4	3.20	4.68	6.78	9.68	13.7
300	2.52	3.69	5.34	7.69	11.0	2.72	3.99	5.77	8.30	11.8	3.03	4.48	6.42	9.22	13.1
350	2.31	3.35	4.97	7.08	10.3	2.45	3.68	5.28	7.74	10.9	2.77	4.09	5.96	8.60	12.3
400	2.06	3.08	4.52	6.57	9.44	2.22	3.32	4.88	7.08	10.2	2.48	3.71	5.43	7.88	11.3
450	1.87	2.83	4.19	6.12	8.84	2.08	3.06	4.54	6.61	9.56	2.26	3.42	5.05	7.36	10.6
500	1.70	2.62	3.91	5.74	8.38	1.88	2.88	4.22	6.20	8.99	2.06	3.16	4.79	6.91	10.0
600	1.42	2.25	3.44	5.11	7.46	1.53	2.43	3.71	5.52	8.17	1.72	2.72	4.14	6.15	8.98
700	1.17	1.95	3.04	4.60	6.79	1.25	2.10	3.29	4.96	7.34	1.42	2.35	3.67	5.58	8.17
800	.95	1.68	2.71	4.16	6.22	1.03	1.81	2.92	4.49	6.71	1.17	1.99	3.17	4.92	7.47
900	.74	1.48	2.40	3.77	5.21	.81	1.45	2.60	4.08	5.18	.93	1.75	2.11	4.56	6.87
1000	.56	1.21	2.18	3.48	5.07	.61	1.31	2.31	3.71	5.60	.71	1.49	2.59	4.15	6.35

L	B.P. 100.3 LBS D=.2617					B.P. 125.3 LBS D=.3147				
	DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
10	25.	35.4	50.1	70.9	100.	27.4	38.8	54.9	77.6	109.
20	20.9	29.6	41.9	59.4	84.	22.9	32.5	46.	65.1	92.1
30	18.4	26.	35.9	52.2	73.9	20.1	28.5	40.4	57.2	81.
40	16.6	23.5	33.3	47.1	66.8	18.2	25.9	36.5	51.9	73.1
50	15.2	21.5	30.5	43.2	61.2	16.7	23.6	33.5	47.4	67.1
60	14.	20.	28.3	40.2	56.8	15.4	21.9	31.	44.1	62.4
70	13.2	18.9	26.6	37.7	53.4	14.4	20.5	29.1	41.3	58.5
80	12.6	17.7	25.1	35.6	50.4	13.5	19.4	27.5	39.	55.2
90	11.8	16.8	23.8	33.8	47.9	13.	18.4	26.1	37.1	52.5
100	11.2	16.	22.7	32.2	45.7	12.4	17.7	25.	35.4	50.2
120	10.3	14.7	20.9	29.7	42.1	11.3	16.1	22.9	32.6	46.2
140	9.61	13.7	19.5	27.6	39.9	10.5	15.	21.3	30.3	43.
160	8.98	12.8	18.2	25.9	36.9	9.86	14.	20.	28.5	40.4
180	8.55	12.4	17.2	25.2	34.9	9.28	13.2	18.9	26.9	38.2
200	7.99	11.4	16.3	23.3	33.1	8.77	12.5	17.9	25.5	36.1
225	7.49	10.9	15.4	22.4	31.8	8.23	11.8	16.9	24.1	34.3
250	7.05	10.1	14.6	20.8	29.7	7.75	11.1	16.	22.8	32.5
275	6.67	9.56	13.9	19.8	28.3	7.33	10.6	15.1	21.8	31.
300	6.32	9.18	13.2	18.9	27.1	6.95	10.1	14.5	20.8	29.7
350	5.82	8.39	12.8	17.4	25.4	6.31	9.26	13.3	19.2	27.4
400	5.28	7.73	11.2	16.2	23.3	5.77	8.51	12.3	17.8	25.6
450	4.80	7.16	10.5	15.2	21.9	5.31	7.89	11.5	16.7	24.
500	4.42	6.65	9.84	14.3	20.6	4.88	7.34	10.8	15.7	22.7
600	3.95	5.80	8.91	12.8	18.6	4.16	6.41	9.59	14.1	20.4
700	3.19	5.06	7.79	11.6	16.9	3.56	5.65	8.59	12.7	18.8
800	2.70	4.48	7.01	10.5	15.6	3.28	5.28	8.25	11.6	17.1
900	2.26	3.94	6.32	9.68	14.4	2.77	4.61	7.22	10.9	16.1
1000	1.86	3.45	5.71	8.89	13.4	2.33	4.08	6.55	10.	15.

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE. TABLE NO 8

INTERNAL DIA. 3.548" C_1 = 3.25 3 1/2" STANDARD PIPE. L_1 = 17' EXTERNAL DIA. 4"

L	B.P 2.3 LBS D = .04352					B.P 5.3 LBS D = .0507					B.P 10.3 LBS D = .06258					B.P 100.3 LBS D = .2617					B.P 125.3 LBS D = .3147				
	DROP IN PRESSURE.					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
10	14.4	20.4	28.8	40.8	57.7	15.5	22.	31.1	44.	62.3	17.3	24.4	34.6	48.9	69.2	35.4	50.	70.8	100.	141.	38.8	54.9	77.6	109.	153.
20	12.3	17.4	24.6	34.9	49.3	13.3	18.8	26.6	37.6	53.2	14.7	20.9	29.5	41.8	59.1	30.2	42.2	60.4	85.2	121.	33.1	46.8	66.3	93.7	132.
30	10.9	15.4	21.8	30.9	43.7	11.8	16.5	23.1	33.4	47.3	13.	18.5	26.2	37.1	52.4	26.7	37.8	53.5	75.8	107.	29.3	41.5	58.8	83.1	117.
40	9.93	14.	19.8	28.	39.7	10.6	15.1	21.4	30.3	42.9	11.8	16.8	23.7	33.6	47.5	24.	34.7	48.6	65.9	97.4	26.6	37.6	53.3	75.3	106.
50	9.12	12.9	18.3	25.8	36.6	9.82	13.9	19.7	27.9	39.4	10.9	15.5	21.9	31.	43.8	22.4	31.7	44.9	62.5	89.9	24.5	34.6	49.2	69.4	98.5
60	8.48	12.	17.	24.1	34.1	9.15	12.2	18.3	26.	36.8	10.1	14.4	20.4	28.9	40.9	20.8	29.5	41.8	59.1	83.6	22.8	32.3	45.9	64.7	91.9
70	7.96	11.2	16.	22.6	32.2	8.60	11.5	17.2	24.4	34.6	9.55	13.5	19.2	27.1	38.4	19.8	27.7	39.7	55.6	78.8	21.5	30.3	43.1	60.8	86.4
80	7.53	10.6	15.1	21.4	30.4	8.14	11.0	16.3	23.1	32.8	9.02	12.8	18.1	25.7	36.4	18.6	26.2	37.2	53.6	74.6	20.3	28.8	40.8	57.6	81.6
90	7.15	10.1	14.4	20.4	28.9	7.72	10.7	15.5	22.	31.1	8.58	12.1	17.2	24.4	34.6	17.5	24.9	35.3	50.	70.9	19.1	27.3	38.5	54.6	77.3
100	6.82	9.69	13.7	19.5	27.6	7.36	10.4	14.8	20.9	29.8	8.19	11.6	16.5	23.3	33.1	16.7	23.8	33.7	47.8	67.7	18.4	26.1	37.1	52.4	74.4
120	6.27	9.43	12.6	19.	25.5	6.78	9.63	13.7	19.4	27.5	7.53	10.7	15.2	21.6	30.5	15.4	21.9	31.1	44.1	62.6	16.9	24.	34.2	48.4	68.7
140	5.84	8.31	11.8	16.7	23.8	6.30	8.97	12.7	18.7	25.7	6.99	9.97	14.1	20.1	28.5	14.3	20.4	29.	41.9	58.4	15.8	22.4	31.9	45.1	64.2
160	5.45	7.79	11.	15.7	22.3	5.90	8.41	11.9	17.	24.1	6.55	9.35	13.3	18.9	26.8	13.4	18.7	27.3	37.7	54.9	14.8	21.	30.	42.4	60.3
180	5.14	7.35	10.5	14.9	21.1	5.54	7.94	11.3	16.1	22.8	6.17	8.83	12.5	17.9	25.4	12.7	18.1	25.8	36.6	52.	13.9	19.9	28.3	40.2	57.1
200	4.89	6.98	9.96	14.1	20.1	5.25	7.53	10.7	15.3	21.7	5.84	8.37	11.9	17.	24.1	12.	17.2	24.5	34.8	49.3	13.2	18.8	26.9	38.2	54.3
225	4.57	6.57	9.39	13.4	19.	4.93	7.09	10.1	14.4	20.5	5.51	7.88	11.3	16.	22.9	11.3	16.4	23.1	32.9	46.8	12.4	17.7	25.3	36.	51.3
250	4.39	6.21	8.90	12.7	18.	4.65	6.70	9.60	13.7	19.5	5.19	7.46	10.7	15.2	21.7	10.7	15.3	21.9	31.3	44.7	11.8	16.8	24.1	34.2	48.8
275	4.08	5.89	8.47	12.1	17.2	4.40	6.37	9.14	13.	18.6	4.91	7.09	10.1	14.5	20.7	10.2	14.6	20.9	29.8	42.4	11.2	16.	23.	32.7	46.6
300	3.88	5.62	8.10	11.5	16.5	4.18	6.07	8.72	12.5	17.8	4.66	6.75	9.71	13.9	19.8	9.67	13.9	20.	28.5	40.6	10.6	15.3	22.4	31.8	44.7
350	3.52	5.14	7.44	10.6	15.2	3.81	5.56	8.04	11.5	16.6	4.14	6.18	8.93	12.8	18.3	8.62	13.4	18.4	26.4	37.6	9.74	13.9	20.3	28.5	41.3
400	3.22	4.85	6.90	9.93	14.2	3.49	5.18	7.45	10.7	15.3	3.88	5.70	8.23	11.9	17.	8.02	12.7	17.	24.5	35.	8.93	13.	18.8	26.9	38.6
450	2.96	4.41	6.48	9.32	13.3	3.26	4.76	6.95	10.	14.4	3.57	5.30	7.73	11.1	16.	7.52	11.	16.	23.	33.	8.31	12.1	17.6	25.3	36.3
500	2.73	4.10	6.03	8.77	12.6	2.96	4.42	6.52	9.47	13.6	3.30	4.94	7.26	10.5	15.1	6.98	10.3	15.	21.7	31.2	7.72	11.3	16.6	23.9	34.3
600	2.34	3.60	5.37	7.87	11.4	2.54	3.89	5.80	8.50	12.3	2.84	4.33	6.46	9.45	13.7	6.08	9.15	13.5	19.6	28.3	6.74	10.	14.8	21.5	31.1
700	2.09	3.18	4.97	7.14	10.7	2.19	3.44	5.21	7.72	11.2	2.44	3.84	5.80	8.51	12.5	5.33	8.17	12.2	17.8	25.9	5.93	9.02	13.4	19.6	28.5
800	1.73	2.82	4.36	6.58	9.61	1.88	3.05	4.71	7.06	10.3	2.10	3.40	5.25	7.85	11.5	4.67	7.34	11.1	16.4	23.9	5.22	8.12	12.3	18.	26.4
900	1.47	2.50	3.95	6.	8.91	1.60	2.71	4.27	6.49	9.61	1.80	3.02	4.77	7.22	10.7	4.09	6.69	10.1	15.2	22.3	4.64	7.38	11.8	16.7	24.6
1000	1.25	2.22	3.58	5.56	8.29	1.34	2.40	3.88	5.99	8.96	1.51	2.19	4.38	6.88	9.96	3.57	5.96	9.54	14.1	20.8	4.03	6.63	10.3	15.5	23.

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE. TABLE NO 9

INTERNAL DIA. 4.026" C_1=3.2 4" STANDARD PIPE. L_1=20' EXTERNAL DIA. 4.5"

L	BOILER PRESSURE 2.3 LBS DROP IN PRESSURE					BOILER PRESSURE 5.3 LBS DROP IN PRESSURE					BOILER PRESSURE 10.3 LBS DROP IN PRESSURE					BOILER PRESSURE 100.3 LBS DROP IN PRESSURE					BOILER PRESSURE 125.3 DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
10	19.4	27.4	38.8	54.9	77.6	20.9	29.6	41.9	59.8	83.8	23.2	32.9	46.5	65.8	93.	47.5	67.3	95.1	134.	190.	52.2	73.8	104.	147.	208.
20	16.8	23.7	33.6	47.5	67.2	18.1	25.6	36.3	51.3	72.6	20.1	28.5	40.3	57.	80.6	41.	58.3	82.4	116.	164.	45.2	63.9	90.4	127.	180.
30	15.	21.2	30.	42.5	60.1	16.2	22.9	32.4	45.8	64.9	17.9	25.4	36.	50.9	72.	36.8	52.1	73.7	104.	147.	40.4	57.2	80.9	114.	161.
40	13.7	19.3	27.4	38.8	54.9	14.7	20.9	29.6	41.8	59.2	16.4	23.2	32.8	46.5	65.7	33.5	47.5	67.2	95.1	134.	36.8	52.	73.7	104.	147.
50	12.6	17.9	25.3	35.9	50.8	13.6	19.3	27.3	38.7	54.8	15.1	21.5	30.4	43.	60.8	31.	43.9	62.2	88.	124.	34.1	48.2	68.3	96.6	136.
60	11.8	16.7	23.7	33.5	47.5	12.7	18.	25.6	36.2	51.3	14.1	20.	28.4	40.2	56.9	29.	41.1	58.2	82.3	116.	31.8	45.1	63.8	90.3	127.
70	11.1	15.7	22.3	31.6	44.7	12.	17.	24.1	34.1	48.3	13.3	18.9	26.8	37.9	53.7	27.4	38.7	54.9	77.1	109.	29.9	42.4	60.	85.	120.
80	10.5	14.9	21.1	29.9	42.4	11.3	16.1	22.8	32.3	45.8	12.6	17.9	25.3	35.9	50.8	25.8	36.6	51.9	73.5	104.	28.4	40.3	56.9	80.7	114.
90	10.	14.2	20.2	28.5	40.4	10.8	15.3	21.8	30.8	43.7	12.	17.	24.1	34.2	48.4	24.6	34.9	49.5	70.1	99.3	27.	38.4	54.3	77.1	108.
100	9.60	13.1	19.3	26.3	38.7	10.3	14.6	20.8	29.4	41.8	11.5	16.3	23.1	32.7	46.4	23.6	33.4	47.4	67.1	95.	25.8	36.7	51.9	73.6	104.
120	8.84	12.5	17.8	25.2	35.8	9.56	13.5	19.2	27.2	38.6	10.6	15.	21.3	30.3	42.9	21.8	30.9	43.8	62.1	87.9	23.8	33.9	48.	68.1	96.5
140	8.24	11.7	16.6	23.6	33.4	8.90	12.6	17.9	25.4	36.1	9.89	14.	19.9	28.3	40.1	20.3	28.8	40.9	58.	82.1	22.3	31.6	44.9	63.6	90.1
160	7.75	11.	15.6	22.2	31.5	8.37	11.9	16.9	23.9	34.	9.29	13.2	18.7	26.6	37.8	19.1	27.2	38.5	54.6	77.4	20.9	29.7	42.2	59.9	84.8
180	7.31	10.4	14.8	21.	29.8	7.91	11.2	16.	22.7	32.2	8.77	12.5	17.7	25.2	35.8	18.	25.6	36.4	51.7	73.3	19.8	28.2	40.	56.8	80.5
200	6.94	9.91	14.1	20.	28.4	7.50	10.7	15.2	21.6	30.7	8.33	11.9	16.9	24.	34.1	17.1	24.4	34.7	49.3	69.9	18.5	26.9	38.1	54.1	76.7
225	6.54	9.35	13.3	18.9	26.9	7.06	10.	14.1	20.4	29.	7.83	11.2	15.9	22.7	32.2	16.1	23.	32.8	46.6	66.1	17.7	25.3	36.	51.1	72.6
250	6.19	8.87	12.6	18.	25.6	6.69	9.57	13.6	19.4	27.6	7.43	10.6	15.2	21.6	30.7	15.3	21.9	31.2	44.4	62.9	16.8	24.	34.2	48.7	69.
275	5.88	8.44	12.	17.2	24.4	6.35	9.11	13.	18.5	26.4	7.08	10.1	14.5	20.7	29.4	14.5	20.8	29.7	42.3	60.1	15.7	22.9	32.1	46.4	65.
300	5.60	8.07	11.5	16.4	23.4	6.06	8.71	12.4	17.7	25.2	6.73	9.68	13.8	19.7	28.1	13.8	20.	28.5	40.6	57.6	15.3	21.9	31.3	44.5	63.3
350	5.12	7.42	10.6	15.2	21.7	5.55	8.13	11.5	16.7	23.4	6.17	8.91	12.8	18.2	26.	12.5	18.3	26.4	37.5	53.4	14.	20.4	28.9	41.2	58.7
400	4.74	6.89	9.93	14.2	20.3	5.12	7.43	10.7	15.3	21.9	5.69	8.27	11.9	17.	24.3	11.7	17.1	24.5	35.1	50.	13.	18.3	27.	37.5	54.9
450	4.39	6.43	9.30	13.2	19.1	4.75	6.94	10.	14.4	20.6	5.29	7.78	11.1	16.	22.9	11.	16.	23.	33.	47.1	12.1	17.	25.3	36.4	51.9
500	4.09	6.03	8.75	12.8	17.9	4.43	6.51	9.67	13.8	19.5	4.93	7.25	10.5	15.1	21.7	10.3	15.	21.7	31.2	44.6	11.3	16.5	23.9	34.2	49.
600	3.59	5.36	7.98	11.4	16.4	3.89	5.80	8.50	12.3	17.7	4.33	6.45	9.45	13.7	19.7	8.90	13.5	19.1	28.3	40.6	10.1	14.8	21.6	31.	44.6
700	3.16	4.81	7.12	10.4	15.	3.43	5.28	7.74	11.4	16.2	3.83	5.80	8.58	12.5	18.1	8.18	12.2	17.9	26.5	37.3	9.	13.4	19.6	28.4	41.
800	2.80	4.34	6.52	9.59	13.9	3.04	4.76	7.05	10.5	15.	3.40	5.25	7.86	11.5	16.7	7.35	11.1	16.4	24.	36.4	8.08	12.7	18.	26.3	38.
900	2.48	3.93	5.99	8.89	13.	2.66	4.26	6.48	9.61	14.	3.01	4.75	7.22	10.7	15.6	6.62	10.2	15.2	22.2	32.4	7.27	11.1	16.7	24.5	35.6
1000	2.19	3.56	5.52	8.27	12.2	2.38	3.88	5.98	8.97	13.1	2.67	4.33	6.66	9.98	14.6	5.96	9.35	14.1	20.9	30.4	6.57	10.2	15.5	22.9	33.4
1500	1.01	2.15	3.74	6.01	9.19	1.18	2.35	4.08	6.51	9.97	1.29	2.64	4.56	7.26	11.1	3.38	6.15	10.	15.6	23.4	3.73	6.77	10.	17.1	26.7
2000		1.47	2.45	4.42	7.19		1.69	3.69	5.60	7.81		1.39	3.03	5.40	8.68	1.40	3.80	7.20	12.4	18.8	1.66	4.20	7.93	13.2	20.6
2500			1.09	3.15	5.63			1.55	3.45	6.12			1.76	3.89	6.82		1.85	4.50	8.20	14.5		1.75	5.40	9.50	16.8
3000				2.06	4.32				2.28	4.73				2.60	5.32			2.75	6.82	12.1			3.21	7.45	13.5
4000					2.24					2.40					2.76				2.88	7.42				3.09	8.45

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS, PER MINUTE, TABLE NO 10

INTERNAL DIA. 5.047" — C_1=3.15 — 5" STANDARD PIPE. — L_1=28' — EXTERNAL DIA. 5.5635"

L	Boiler Pressure 2.3 lbs — Drop in Pressure 1 oz	2 oz	4 oz	8 oz	16 oz	Boiler Pressure 5.3 lbs — Drop in Pressure 1 oz	2 oz	4 oz	8 oz	16 oz
10	31.9	45.1	63.8	90.3	127.	34.4	48.6	68.8	97.2	137
20	28.3	40.1	56.7	80.2	113.	30.6	43.3	61.3	86.6	122
30	25.8	36.5	51.6	72.9	103.	27.8	39.3	55.7	78.7	111
40	23.8	33.7	47.7	67.4	95.4	25.7	36.4	51.4	72.8	103
50	22.1	31.4	44.4	62.8	88.9	23.9	33.9	47.9	67.9	96
60	21.1	29.5	42.3	59.1	84.7	22.5	32.2	45.1	64.5	90.4
70	19.8	28.	39.6	56.1	79.3	21.3	29.9	42.8	59.9	85.7
80	18.8	26.6	37.7	53.4	75.5	20.	28.6	40.2	57.4	91.5
90	18.	25.5	36.1	51.1	72.3	19.4	27.5	39.	55.2	78.1
100	17.2	24.4	34.6	49.	69.4	18.6	26.3	37.4	52.7	74.9
120	16.	22.7	32.2	45.6	64.5	17.2	24.4	34.7	49.1	69.6
140	15.	21.3	30.2	42.7	60.5	16.1	22.9	32.5	46.1	65.3
160	14.1	20.1	28.6	40.4	57.2	15.2	21.7	30.7	43.6	61.7
180	13.4	19.	27.	38.3	54.3	14.5	20.6	29.2	41.4	58.6
200	12.8	18.2	25.8	36.6	51.8	13.7	19.5	27.8	39.4	55.9
225	12.	17.1	24.3	34.6	49.9	13.	18.5	26.4	37.4	53.1
250	11.4	16.3	23.2	32.9	46.8	12.3	17.6	25.1	35.6	50.5
275	10.9	15.5	22.2	31.5	44.8	11.8	16.8	24.	34.1	48.4
300	10.4	14.9	21.2	30.2	43.	11.3	16.2	23.	32.8	46.4
350	9.65	13.8	19.7	28.1	39.9	10.4	14.9	21.3	30.4	43.1
400	8.94	12.6	18.4	26.2	37.4	9.68	13.9	19.9	28.4	40.4
450	8.38	12.1	17.3	24.8	35.4	9.06	13.	18.7	26.8	38.2
500	7.88	11.4	16.5	23.1	33.5	8.51	12.3	17.7	25.4	36.2
600	7.04	10.2	14.9	21.4	30.6	7.61	11.1	16.1	23.1	33.
700	6.33	9.76	13.6	19.7	28.2	6.85	10.1	14.7	21.2	30.4
800	5.75	8.58	12.5	18.2	26.2	6.21	9.28	13.5	19.7	28.3
900	5.23	7.90	11.6	16.8	24.4	5.67	8.56	12.6	18.4	26.6
1000	4.77	7.31	10.9	15.9	23.2	5.17	7.92	11.8	17.3	25.
1500	2.99	5.07	8.02	12.1	18.	3.26	5.51	8.69	13.2	19.5
2000	1.64	3.45	6.	9.62	14.7	1.81	3.76	6.52	10.4	15.9
2500		2.35	4.42	8.10	12.2		2.83	4.82	8.32	13.2
3000			3.07	5.02	10.2		1.10	3.36	6.56	11.
4000				2.82	[illegible]				3.67	7.58

L	Boiler Pressure 10.3 lbs — Drop in Pressure 1 oz	2 oz	4 oz	8 oz	16 oz	Boiler Pressure 100.3 lbs — Drop in Pressure 1 oz	2 oz	4 oz	8 oz	16 oz
10	38.2	54.1	76.4	108	153	78.2	110	156	221	313
20	34.	48.2	68.	96.4	136	69.6	98.5	139	197	278
30	30.9	43.7	61.9	87.5	123	63.2	89.5	126	179	253
40	28.6	40.4	57.2	80.9	114	58.4	82.6	117	165	234
50	26.5	37.7	53.2	75.5	106	54.5	77.1	109	154	218
60	25.	35.5	50.1	71.	100	51.3	72.6	102	145	205
70	23.7	33.6	47.5	67.3	95.2	48.5	68.7	97.2	137	194
80	22.5	32.	45.2	64.1	90.5	46.2	65.4	92.6	131	185
90	21.5	30.5	43.2	61.3	86.6	44.2	62.6	88.6	125	177
100	20.6	29.3	41.4	58.8	83.1	42.4	60.	85.	120	170
120	19.2	27.3	38.5	54.7	77.2	39.3	55.8	78.9	112	158
140	17.9	25.5	36.1	51.3	72.5	36.7	52.2	73.9	105	148
160	16.9	24.1	34.1	48.4	68.5	34.7	49.8	69.9	99.1	140
180	16.	22.8	32.2	46.	65.	33.	46.8	66.4	94.	133
200	15.3	21.8	30.9	43.9	62.1	31.4	44.7	63.3	89.9	127
225	14.4	20.6	29.3	41.6	58.9	29.7	42.2	60.	85.	120
250	13.7	19.6	27.9	39.6	56.1	28.3	40.7	57.1	82.1	115
275	13.1	18.7	26.6	37.9	53.7	27.	38.5	54.7	77.7	110
300	12.5	17.9	25.5	36.3	51.6	25.8	36.9	52.4	74.5	106
350	11.5	16.6	23.7	33.8	47.9	23.8	34.1	48.6	69.2	98.2
400	10.7	15.5	22.1	31.7	44.9	22.3	31.9	45.6	64.9	92.2
450	10.	14.5	20.8	29.8	42.4	20.8	30.	42.9	61.2	87.
500	9.48	13.2	19.7	27.2	40.2	19.6	28.3	40.6	58.	82.6
600	8.96	12.4	17.9	25.7	36.7	18.7	25.6	36.9	52.9	75.4
700	7.63	11.2	16.3	23.6	33.8	16.	23.4	33.9	48.7	69.6
800	6.94	10.4	15.1	22.2	31.5	14.7	21.6	31.5	45.3	65.
900	6.33	9.50	14.	20.4	29.5	13.5	20.	29.3	42.4	61.
1000	5.58	8.84	12.7	19.2	27.	12.5	18.7	27.6	40.	57.9
1500	3.68	6.19	9.71	14.7	21.7	8.44	13.5	20.9	31.	45.5
2000	2.09	4.27	7.33	11.7	17.8	5.51	9.94	17.2	25.	37.6
2500		2.72	5.45	9.36	14.8	3.09	7.06	12.6	20.6	31.8
3000			3.86	7.44	12.4		4.61	9.74	17.	27.2
4000			1.19	4.24	8.57			4.80	11.1	20.

L	Boiler Pressure 125.3 lbs — Drop in Pressure 1 oz	2 oz	4 oz	8 oz	16 oz
10	85.7	121	171	242	343
20	76.3	108	152	216	305
30	69.4	98.7	139	196	278
40	64.	90.6	128	181	256
50	59.8	84.6	119	169	239
60	56.1	79.6	112	159	225
70	53.2	75.4	106	151	213
80	50.6	71.8	101	144	203
90	48.4	68.6	97.1	137	194
100	46.4	65.8	93.2	132	187
120	43.1	61.2	86.6	123	178
140	40.4	57.3	81.2	115	163
160	38.1	54.2	76.7	109	154
180	36.1	51.3	72.6	108	146
200	34.4	49.	69.4	98.6	139
225	32.6	46.4	65.9	94.4	132
250	31.	44.1	62.7	88.9	126
275	29.6	42.2	60.	85.2	120
300	28.4	40.5	57.6	81.8	116
350	26.1	37.5	53.3	76.	108
400	24.4	35.	50.	71.2	101
450	23.	33.	47.2	67.2	95.6
500	21.6	31.2	44.6	63.7	90.7
600	19.4	28.1	40.5	58.	82.7
700	17.7	25.8	37.3	53.5	76.5
800	16.1	23.8	34.5	49.8	71.3
900	14.8	22.	32.2	46.6	66.9
1000	13.7	20.5	30.2	43.9	63.2
1500	9.13	14.9	22.8	34.1	49.9
2000	6.13	11.	17.8	27.6	41.3
2500	3.51	7.88	14.	22.7	35.
3000	1.21	5.20	10.8	18.8	30.
4000			4.97	12.4	22.1

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE.

TABLE NO 11

INTERNAL DIA. 6.065" $C_1 = 3.1$ **6" STANDARD PIPE.** $L_1 = 36'$ EXTERNAL DIA 6.6256"

L	BOILER PRESSURE 2.3 LBS					BOILER PRESSURE 5.3 LBS					BOILER PRESSURE 10.3 LBS					BOILER PRESSURE 100.3 LBS					BOILER PRESSURE 125.3 LBS				
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
10	47.6	67	95.2	134	190	51.3	72.7	102	145	205	57.	80.7	114	161	228	116	165	238	330	466	128	181	256	362	512
20	43.1	61	86.2	122	172	46.6	65.8	93.2	131	186	51.8	73.1	103	146	207	105	149	211	299	423	116	164	232	328	464
30	39.7	56.1	79.5	112	159	42.9	60.7	85.8	121	171	47.7	67.4	95.4	134	190	97.4	137	195	275	390	106	151	212	302	427
40	36.9	52.2	73.9	104	148	40.	56.6	80.	118	160	44.4	62.7	88.9	125	177	90.6	128	181	257	363	99.5	141	199	282	398
50	34.8	49.1	69.6	98.3	139	37.5	53.	75.1	106	150	41.6	58.9	83.6	118	167	85.1	120	170	241	341	93.6	132	187	265	375
60	32.8	46.5	65.8	93.1	131	35.4	50.2	71.	100	142	39.4	55.8	79.	111	158	80.5	114	161	228	323	88.5	125	177	250	356
70	31.2	44.2	62.5	88.6	125	33.7	47.7	67.6	95.6	135	37.5	53.	75.1	106	150	76.7	108	153	217	307	84.	119	168	238	337
80	29.8	42.3	59.8	84.7	119	32.2	45.7	64.6	91.5	129	35.8	50.7	71.8	101	143	73.3	104	147	208	294	82.7	114	166	228	322
90	28.6	40.5	57.4	81.2	115	30.9	43.8	62.	87.7	124	34.4	48.6	68.9	97.3	138	70.3	99.5	141	199	282	77.2	109	155	219	310
100	27.5	39.	55.2	78.2	110	29.8	42.2	59.7	84.6	119	32.9	46.7	66.1	93.6	132	67.5	95.7	135	192	271	74.2	105	149	210	298
120	25.6	36.3	51.4	73.	103	27.7	39.7	55.6	79.7	111	30.8	43.6	61.8	87.5	124	63.1	89.3	125	179	253	69.1	98.	139	196	278
140	24.1	34.2	48.4	68.5	97.	26.6	36.9	52.4	74.1	105	28.9	41.	58.1	82.3	116	59.4	84.	119	168	238	64.9	92.1	130	185	261
160	22.8	32.4	45.9	65.	92.	24.7	35.	49.6	70.2	99.4	27.3	38.8	55.	77.9	110	55.9	79.6	112	159	225	61.5	87.2	124	175	248
180	21.7	30.8	43.7	61.9	87.6	23.5	33.3	47.2	66.9	94.7	26.	37.	52.3	74.4	105	53.3	75.7	107	152	215	58.7	82.9	118	166	236
200	20.7	29.4	41.7	59.1	83.7	22.4	31.9	45.1	63.9	90.5	24.9	35.3	50.1	70.9	100	51.	72.3	103	145	206	55.8	79.3	112	159	225
225	19.6	27.9	39.6	56.2	79.6	21.3	30.2	42.9	60.7	86.1	23.5	33.4	47.5	67.3	95.4	48.4	68.9	97.4	138	195	52.9	75.2	106	151	214
250	18.7	26.6	37.8	53.6	76.	20.2	28.7	40.8	57.9	82.1	22.4	31.9	45.3	64.3	91.1	46.1	65.5	93.	132	187	50.6	71.8	102	144	205
275	17.9	25.5	36.2	51.4	72.8	19.3	27.5	39.1	55.5	78.7	21.4	30.5	43.3	61.6	87.2	44.1	62.8	89.	126	179	48.3	68.8	97.6	138	196
300	17.2	24.5	34.8	49.4	70.	18.6	26.5	37.6	53.4	75.7	20.6	29.3	41.7	59.2	84.	42.3	60.3	85.5	121	172	46.4	65.9	93.9	133	189
350	15.9	22.7	32.3	46.	65.2	17.2	24.6	35.	49.7	70.5	19.1	27.2	38.8	55.1	78.2	39.3	56	79.6	113	160	43.2	61.3	87.7	124	175
400	14.8	20.7	30.3	43.1	61.2	16.1	23.2	32.8	47.1	66.2	17.8	25.5	36.4	51.7	73.5	36.7	52.5	74.6	106	150	40.1	57.4	81.7	116	165
450	13.5	20.1	27.6	40.8	57.9	15.1	21.6	30.9	44.	62.5	16.8	24.1	34.4	49.	69.5	34.7	49.5	70.6	100	142	37.7	54.2	77.1	110	156
500	12.7	19.	27.1	38.7	55.	14.2	20.4	29.2	41.7	59.3	15.9	22.8	32.5	46.5	66.	31.7	46.8	64.9	95.1	135	35.8	51.3	73.8	104	148
600	11.9	17.3	24.8	35.4	50.3	12.9	18.6	26.7	38.2	54.3	14.3	20.7	29.6	42.4	60.3	29.6	42.8	61.1	87.2	124	32.3	46.6	66.7	95.3	135
700	10.8	15.8	22.7	32.6	46.5	11.7	17.1	24.5	35.3	50.2	13.	18.9	27.3	38.1	56.	27.2	39.3	56.1	80.6	114	29.6	42.9	61.6	88.2	126
800	9.97	14.6	21.1	30.4	43.4	10.8	15.8	22.9	32.9	47.	12.	17.6	25.4	36.6	51.2	25.1	36.5	52.5	75.2	107	27.2	39.7	57.3	82.2	117
900	9.18	12.8	19.8	28.5	41.9	10.	14.7	21.4	30.4	44.2	11.1	16.4	23.8	34.3	49.1	23.3	34.	49.2	70.6	101	25.2	37.	53.6	77.1	110
1000	8.53	11.7	18.5	26.9	39.6	9.36	13.8	20.1	28.1	41.8	10.3	15.3	22.3	32.3	46.4	21.7	32.1	46.3	67.	95.5	23.7	34.6	50.9	72.8	105
1500	6.28	9.45	14.7	21.1	31.7	6.57	10.1	15.4	22.6	33.2	7.30	11.4	17.2	25.4	37.	16.	24.1	36.2	52.9	76.6	16.9	26.4	39.1	58.	83.4
2000	4.21	7.18	11.3	17.3	25.6	4.67	7.86	12.4	18.8	27.9	5.28	8.74	13.8	20.8	31.	11.9	19.2	29.5	44.1	64.6	12.2	20.2	31.5	47.5	70.
2500	2.74	5.40	9.15	14.4	22.	3.06	5.91	10.	15.7	23.9	3.20	6.63	11.1	17.2	26.5	8.70	15.2	24.4	37.5	55.9	8.20	15.6	25.2	40.	59.2
3000	1.65	3.97	7.62	12.2	19.4	1.72	4.33	8.07	13.8	20.7	1.82	4.84	9.90	14.7	23.	4.90	10.9	19.8	32.2	48.8	5.20	11.8	21.	34.1	52.5
4000		1.31	4.27	8.50	14.4		1.60	4.81	9.36	15.8		1.82	5.30	10.4	17.5	1.20	4.40	11.7	24.	38.6	1.40	5.30	13.3	24.7	40.6

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE.

TABLE NO 12

7" STANDARD PIPE.

INTERNAL DIA. 7.023" — C_1 = 3.025 — L_1 = 44' — EXTERNAL DIA. 7.6257"

L	B.P. 2.3 LBS D = .04352					B.P. 5.3 LBS D = .0507					B.P. 10.3 LBS D = .06253					B.P. 100.3 LBS D = .2617					B.P. 125.3 LBS D = .3147				
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
10	65.1	92.1	130	184	260	70.8	99.4	140	190	281	78.0	110	156	221	312	159	226	319	453	638	175	247	350	495	700
20	59.8	84.6	119	169	239	64.5	91.2	129	182	256	71.7	101	143	203	287	146	207	293	415	586	160	227	321	455	643
30	55.6	78.7	111	157	222	60.0	84.9	120	169	240	66.7	94.2	133	188	267	138	192	277	385	544	149	211	299	423	598
40	52.3	73.8	104	147	209	56.4	79.7	112	159	225	62.5	88.4	125	177	250	127	181	255	362	511	140	198	281	397	562
50	49.2	69.6	98.6	139	197	53.2	75.2	106	150	213	59.1	83.6	118	167	236	120	171	241	342	483	132	187	265	375	531
60	46.8	66.2	93.7	132	187	50.5	71.0	101	143	202	56.6	79.4	112	159	225	115	162	230	325	460	126	178	252	357	504
70	44.7	63.3	89.5	126	179	48.2	68.3	96.6	136	193	53.6	75.9	107	152	215	109	155	219	310	439	120	170	241	341	482
80	42.8	60.7	85.8	121	172	46.2	65.5	92.6	131	185	51.4	72.8	103	145	206	105	148	210	297	421	115	163	231	326	462
90	41.2	58.3	82.6	116	165	44.5	63.0	89.1	126	178	49.3	70.0	99.0	140	198	101	143	202	286	405	111	157	222	314	445
100	39.7	56.3	79.6	112	159	42.9	60.7	86.	121	172	45.1	67.4	90.4	135	191	97.4	138	195	276	391	107	151	214	303	429
120	37.1	52.6	74.5	105	149	40.1	56.8	80.4	114	161	44.6	63.2	89.4	126	179	91.2	129	183	258	366	100	142	201	284	402
140	35.0	49.7	70.3	99.6	141	37.8	53.6	75.9	107	152	42.1	59.6	84.4	119	169	86.0	122	172	244	345	94.4	134	189	268	379
160	33.3	47.1	66.7	94.5	133	35.8	51.3	72.0	102	144	39.8	56.5	80.0	113	160	81.6	116	164	232	328	89.6	127	180	254	360
180	31.6	44.9	63.6	90.1	127	34.2	48.5	68.7	97.3	138	38.0	53.8	76.4	108	153	77.7	110	156	221	313	85.4	120	171	242	343
200	30.3	43.0	60.9	86.3	122	33.2	46.5	65.8	93.3	132	36.3	51.5	73.0	103	146	74.5	105	149	212	300	81.6	116	164	232	329
225	29.3	40.8	58.0	82.0	116	31.1	43.9	62.6	88.7	125	34.5	49.0	69.5	98.5	139	70.8	100	142	201	285	77.7	115	156	221	313
250	27.5	39.0	55.4	78.5	111	29.6	42.2	59.8	84.8	120	32.9	46.8	66.4	94.1	133	67.6	95.4	136	191	273	74.2	105	149	211	300
275	26.4	37.5	53.2	75.4	107	28.4	40.4	57.3	81.3	115	31.6	44.9	63.7	90.4	128	64.9	91.5	130	185	262	71.2	101	143	203	287
300	25.3	36.0	51.1	72.5	103	27.8	38.9	55.1	78.3	111	30.3	43.1	61.2	86.8	123	62.3	88.5	125	178	252	68.4	97.1	137	195	276
350	23.5	33.5	47.6	67.6	95.8	25.4	36.2	51.5	73.0	104	28.3	40.3	57.2	81.3	115	58.0	82.4	117	166	235	63.6	90.4	128	182	258
400	22.1	31.5	44.8	63.6	90.3	23.7	34.0	48.2	68.7	97.2	26.5	37.8	53.7	76.4	108	54.3	77.4	110	156	221	59.7	85.5	121	171	248
450	20.8	29.7	42.4	60.2	85.5	22.4	32.1	45.6	65.0	92.1	25.0	35.7	50.8	72.3	103	51.4	73.2	101	148	210	56.4	80.4	114	162	230
500	19.7	28.2	40.1	57.2	81.3	21.3	30.5	43.5	61.9	87.7	23.7	33.9	48.3	68.7	97.5	48.7	69.5	99.0	140	200	53.5	76.4	108	155	219
600	17.8	25.6	36.7	52.3	74.5	19.2	27.7	39.6	56.5	80.3	21.5	30.8	44.1	62.8	89.3	44.8	63.4	90.6	128	183	48.6	69.6	99.3	141	200
700	16.4	23.7	34.0	48.5	69.1	17.8	25.5	36.6	52.8	74.5	19.7	28.3	40.8	57.9	82.9	40.8	58.6	83.8	119	170	44.8	64.3	92.0	131	186
800	15.1	22.0	31.6	45.3	64.6	16.3	23.7	34.1	48.8	69.7	18.2	26.4	38.0	54.3	77.5	37.8	54.6	78.2	111	159	41.5	59.8	85.8	122	174
900	14.0	20.5	29.6	42.5	60.8	15.2	22.1	32.0	45.9	65.6	16.9	24.7	35.6	51.1	73.0	35.3	51.1	73.5	105	150	38.8	55.1	80.7	115	164
1000	13.2	19.3	28.0	40.2	57.5	14.2	20.9	30.2	43.6	62.2	15.8	23.1	33.6	48.2	69.1	33.4	48.1	70.0	99.5	148	36.7	52.8	76.9	109	157
1500	9.68	14.7	21.8	31.9	46.2	10.5	15.9	23.6	34.5	49.9	11.7	17.8	26.3	38.5	55.5	25.0	37.4	54.8	79.6	114	27.5	41.1	60.2	87.4	125
2000	7.26	11.6	18.8	26.6	39.0	7.96	12.6	19.3	28.7	42.1	8.84	14.1	21.5	32.0	46.9	19.5	30.3	45.4	67.0	97.2	21.4	33.2	49.8	73.4	106
2500	5.33	8.74	14.8	21.6	33.7	5.76	10.0	16.0	24.4	36.5	6.56	11.2	17.9	27.3	40.6	15.2	24.9	38.5	57.8	84.9	18.7	27.1	42.2	63.3	93.3
3000	3.67	7.25	12.3	19.5	29.7	4.01	7.90	13.5	21.1	32.2	4.62	8.93	15.0	23.6	35.8	11.3	20.5	32.9	51.2	75.4	12.8	22.5	36.1	55.5	82.7
4000		5.00	9.42	14.6	23.5	1.09	4.40	9.17	15.9	25.4	1.10	5.09	9.88	17.8	27.5	6.60	13.8	24.1	39.4	61.8	6.3	12.1	26.6	38.2	67.2

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE. TABLE NO 13

INTERNAL DIA. 7.981" — $C_1 = 2.96$ — 8" STANDARD PIPE — $L_1 = 52$ — EXTERNAL DIA. 8.625

L	B.P. 2.3 LBS D=.04352					B.P 5.3 LBS D=.0507					B.P 10.3 LBS D=.06253					B.P. 100.3 LBS D=.2617					B.P. 125.3 LBS D=.3147				
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
10	85.3	121	171	242	342	92.3	130	184	261	369	102	145	205	290	410	209	296	419	593	839	230	325	460	650	921
20	79.4	112	158	224	317	85.7	121	171	242	342	95.1	134	190	269	380	194	275	389	550	778	218	302	427	604	854
30	74.3	105	148	210	297	80.3	113	160	227	321	89.2	126	178	252	356	182	258	365	516	730	200	282	400	565	800
40	70.2	99.4	140	198	280	75.8	107	151	214	303	84.2	119	168	238	336	172	243	345	487	689	188	267	377	534	755
50	66.6	94.0	133	188	266	71.9	101	143	203	287	79.8	113	159	226	319	163	231	327	462	654	178	253	357	506	717
60	63.5	89.9	127	180	254	68.6	97.0	137	194	274	76.2	107	152	215	305	156	220	312	441	624	171	242	342	484	685
70	60.9	86.1	122	172	244	65.7	92.9	131	186	263	73.0	103	146	206	292	149	211	299	422	598	164	231	328	463	656
80	58.5	82.8	117	165	234	63.1	89.4	126	179	253	70.2	99.2	140	198	281	143	203	287	406	575	157	223	315	446	630
90	56.4	79.9	113	160	226	60.9	86.1	122	172	244	67.5	95.6	135	191	271	138	196	277	392	554	152	215	304	430	608
100	54.5	77.2	109	154	218	58.8	83.2	117	166	235	63.5	92.4	131	185	262	134	186	268	378	536	146	207	293	415	587
120	51.8	72.4	102	145	205	55.2	78.2	110	156	221	61.4	86.8	123	174	246	125	177	251	355	503	137	195	275	390	551
140	48.4	68.3	97.0	137	194	52.2	73.9	105	148	209	57.9	82.1	116	164	233	118	168	237	336	475	130	189	261	369	522
160	45.9	65.1	92.2	130	185	49.6	70.3	99.5	141	199	55.1	78.1	110	156	221	113	160	226	320	453	124	175	248	351	497
180	43.9	62.8	88.1	125	176	47.4	67.2	95.1	135	190	52.6	74.6	106	150	212	108	153	211	306	433	118	167	237	335	475
200	42.1	59.7	84.5	120	169	45.4	64.4	91.2	129	183	50.4	71.5	101	143	203	103	146	201	293	415	113	160	227	322	455
225	40.0	56.8	80.5	114	161	43.3	61.3	87.0	123	174	48.1	68.2	96.6	137	193	98.4	141	197	282	396	108	153	217	306	434
250	38.3	54.4	77.1	109	154	41.4	58.7	83.2	118	167	45.9	65.2	92.4	131	185	94.1	133	189	268	379	103	146	207	293	416
275	36.8	52.2	74.0	105	148	39.7	56.4	80.0	113	160	44.1	62.6	88.8	126	178	90.3	128	181	257	364	99.	140	199	282	399
300	35.3	50.3	71.2	101	143	38.2	54.3	76.9	109	154	42.4	60.3	85.5	121	171	86.9	123	175	248	351	95.3	135	192	272	385
350	32.9	46.9	66.5	94.4	133	35.6	50.6	71.8	102	144	39.5	56.2	79.7	113	160	81.0	115	163	237	328	88.9	126	179	254	360
400	30.9	44.0	62.6	88.8	126	33.5	47.6	67.6	95.9	136	37.1	52.9	75.1	106	151	76.7	108	154	228	309	83.6	119	169	239	339
450	29.2	41.7	59.3	84.2	119	31.8	45.0	64.0	90.9	129	35.1	50.0	71.1	100	143	72.0	102	145	206	293	79.1	112	160	227	321
500	27.7	39.6	56.4	80.1	113	30.0	42.6	60.9	86.6	123	33.9	47.5	67.7	96.1	136	68.6	97.4	139	196	280	75.1	107	152	216	306
600	25.3	36.3	51.7	73.6	104	27.3	39.1	55.8	79.4	113	30.4	43.5	62.0	88.2	125	62.3	98.1	127	180	256	68.5	97.7	139	198	281
700	23.3	33.5	47.9	68.3	97.2	25.1	36.2	51.6	73.7	105	28.0	40.2	57.5	81.8	116	57.6	82.5	118	167	238	63.2	90.6	129	184	261
800	21.6	31.2	44.7	63.6	90.9	22.8	33.5	48.2	68.9	97.9	26.0	37.5	53.7	76.6	109	53.5	77.0	110	157	223	58.8	84.5	121	172	245
900	20.2	29.2	42.0	60.0	85.7	21.8	31.6	45.3	64.9	92.6	24.3	35.2	50.5	72.2	103	50.1	72.2	103	148	210	55.1	79.3	113	162	231
1000	19.0	27.6	39.9	56.9	81.8	20.5	29.8	42.9	61.5	87.7	22.8	33.2	47.7	68.4	97.4	47.1	68.2	98.0	140	199	51.7	74.8	107	153	219
1500	14.5	21.5	31.6	45.7	65.8	15.6	23.2	34.0	49.4	70.8	17.4	26.3	37.9	55.8	78.9	36.2	53.5	78.1	113	162	40.	58.9	85.9	124	178
2000	11.3	17.5	26.2	38.5	55.9	12.3	18.9	28.8	41.5	60.4	13.7	21.1	31.5	46.3	67.2	28.8	44.0	65.3	95.6	148	31.7	48.3	71.7	105	151
2500	8.9	14.4	22.2	33.3	48.9	9.7	15.6	24.4	36.0	52.9	10.9	17.5	26.9	40.1	59.8	23.7	34.2	55.9	88.0	121	25.6	40.5	61.5	91.2	133
3000	6.8	11.9	19.1	29.2	43.4	7.5	13.0	20.7	31.6	47.0	8.5	14.5	23.1	36.2	52.4	18.4	33.4	48.3	78.2	108	20.3	34.	53.	80.4	108
4000	3.4	7.8	14.0	22.7	35.2	3.9	8.6	15.8	24.7	38.1	4.4	9.7	17.1	27.6	42.5	10.7	21.4	36.6	58.1	88.4	12.	23.8	40.4	64.1	97.3

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE

TABLE NO 14

INTERNAL DIA. 8.941" C = 2.825 9" STANDARD PIPE. L = 60' EXTERNAL DIA. 9.625"

L	B.P. 2.3 LBS. D=.04352					B.P. 5.3 LBS. D .0507					B.P. 10.3 LBS. D=.06253					B.P. 100.3 LBS. D=.2617					B.P. 125.3 LBS. D=.3147				
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.
10	109	154	218	308	436	117	167	235	334	471	130	185	261	370	523	267	378	535	756	1070	298	414	586	829	1173
20	102	143	204	287	408	110	155	220	311	440	122	173	244	345	489	250	354	500	708	1001	274	387	549	775	1098
30	98.6	136	197	272	384	103	146	207	293	415	115	163	230	326	461	235	333	473	667	946	258	365	517	731	1034
40	91.2	129	182	258	365	98.5	139	197	278	394	109	154	218	309	437	223	316	447	632	895	245	347	490	694	980
50	86.9	123	174	246	348	93.8	132	187	265	375	104	147	208	294	417	213	301	426	603	853	233	330	467	661	935
60	83.1	117	166	235	333	89.9	127	180	254	360	99.7	141	199	282	399	204	288	408	577	817	224	316	448	633	896
70	79.9	113	160	226	320	86.7	122	177	244	345	95.8	135	191	271	383	198	276	397	553	785	215	304	430	608	860
80	77.	109	154	218	308	83.1	117	166	235	333	92.3	130	185	261	370	189	267	378	534	756	207	293	414	585	829
90	74.3	105	149	210	298	80.2	113	160	227	321	89.	126	178	252	357	182	258	365	516	731	200	283	400	566	800
100	71.9	101	144	203	288	77.7	110	155	220	311	86.3	122	173	244	346	176	249	353	499	707	193	274	387	548	775
120	67.5	95.9	136	192	272	73.1	103	146	207	293	81.2	115	163	230	326	166	235	333	471	666	183	258	365	517	731
140	64.2	90.	129	182	258	69.3	98.	139	196	278	77.	109	154	218	309	157	223	316	447	632	173	245	346	490	693
160	61.2	86.7	123	174	246	66.	93.	132	187	265	73.3	104	147	208	294	150	213	301	426	603	165	233	330	467	661
180	58.5	82.8	117	166	235	63.2	89.	127	179	254	70.2	99.3	141	199	282	143	203	288	408	577	157	214	316	447	632
200	56.1	79.5	113	160	226	60.5	85.9	122	172	244	67.4	95.4	135	191	271	138	195	276	391	554	151	204	303	429	608
225	53.3	75.9	107	152	215	57.3	81.9	116	164	233	64.3	91.1	129	183	259	131	186	264	374	529	144	196	289	410	580
250	51.3	72.7	103	146	206	55.4	78.5	111	157	223	61.6	87.3	124	175	248	126	179	253	358	508	136	188	277	393	556
275	49.1	69.9	98.8	140	198	53.2	75.4	107	151	214	59.1	83.9	119	168	238	121	172	243	344	488	133	182	267	378	535
300	47.4	67.3	95.5	136	191	51.1	72.6	103	141	207	56.9	80.7	114	162	230	117	165	235	332	471	128	170	257	365	516
350	44.2	62.9	89.3	126	179	47.5	67.9	95.9	136	192	53.2	75.5	107	151	215	109	155	220	311	441	120	160	241	341	483
400	41.7	59.3	84.3	112	169	45.	64.	90.9	129	183	50.1	70.6	101	142	203	103	146	207	293	416	112	152	227	322	455
450	39.5	56.2	79.9	110	160	42.2	60.7	85.8	122	172	47.4	67.4	95.9	135	192	97.4	138	196	278	395	107	144	215	305	432
500	37.5	53.5	76.1	108	153	40.6	57.3	82.2	116	165	45.1	64.2	91.4	129	184	92.7	131	187	265	376	101	133	205	291	412
600	34.4	49.1	69.9	99.3	141	37.1	52.9	75.4	107	152	41.2	58.3	83.3	119	169	84.3	121	172	244	346	93.1	123	189	258	360
700	31.8	45.6	64.9	92.5	131	34.8	49.1	70.	100	141	38.	55.	77.7	110	157	78.6	112	160	227	328	86.1	119	175	249	353
800	29.6	41.	60.7	83.5	123	31.7	46.3	65.	94.3	132	35.5	48.2	72.8	103	147	73.	104	149	212	302	80.4	115	164	233	331
900	27.7	39.9	57.2	81.6	116	30.0	42.	61.3	87.9	125	33.3	47.9	68.6	97.3	139	68.8	98.7	141	201	286	75.6	109	155	220	313
1000	26.1	37.7	54.2	77.3	110	28.2	40.7	58.4	83.4	118	31.4	45.3	65.	92.3	132	64.9	93.4	133	190	271	71.2	102	146	219	307
1500	20.2	29.7	43.2	62.4	89.5	21.5	34.3	46.3	71.7	96.7	24.4	35.7	52.1	74.3	107	51.	74.4	107	154	221	56.	81.7	118	169	242
2000	16.2	24.5	36.3	52.9	76.5	17.6	26.5	39.3	62.2	82.7	19.6	29.5	43.7	63.6	91.9	41.7	62.1	91.	131	189	45.6	68.2	100	144	208
2500	13.2	20.7	31.2	46.2	67.3	14.3	22.4	34.3	49.9	72.7	16.	24.9	37.6	55.5	80.9	34.7	53.1	79.	115	168	38.2	58.4	86.7	127	183
3000	10.7	17.5	27.2	40.3	60.2	11.6	18.9	29.4	44.2	65.	13.	21.2	32.8	48.2	72.4	29.	45.3	69.5	103	150	32.2	50.4	76.2	113	165
4000	6.5	12.9	20.3	32.7	49.4	7.1	13.5	22.6	35.4	53.5	8.1	15.2	25.3	39.5	59.6	19.9	34.4	55.	84.	125	22.0	33.	60.5	92.5	137

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE

TABLE NO 15

INTERNAL DIA 10.02" C_1 = 2.75 10" STANDARD PIPE. L_1 = 69 EXTERNAL DIA. 10.75

L	B.P. 2.3 Lbs D = .04352					B.P. 5.3 Lbs D = .0507					B.P. 10.3 Lbs D = .06253					B.P. 100.3 Lbs D = .2617					B.P. 125.3 Lbs D = .3147"				
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ	1 OZ	2 OZ	4 OZ	8 OZ	16 OZ
10	138	196	277	392	554	149	211	299	423	599	166	235	332	470	665	340	481	680	963	1362	378	527	746	1054	1492
20	130	184	261	369	522	141	199	282	399	565	156	221	313	443	626	320	453	641	907	1282	351	497	702	994	1405
30	123	175	247	350	495	133	189	267	378	535	148	210	297	420	594	304	430	608	860	1216	333	471	666	942	1332
40	118	166	236	333	472	127	180	255	360	510	141	200	283	400	566	289	410	579	820	1158	317	448	634	897	1268
50	113	159	225	319	451	121	177	248	345	487	135	195	271	383	542	277	392	554	785	1108	304	430	608	860	1216
60	108	153	217	306	433	117	165	234	331	468	130	184	260	368	520	266	377	533	754	1066	292	413	584	826	1169
70	104	147	209	295	417	112	159	225	319	451	125	177	250	354	501	256	362	512	725	1025	281	397	562	795	1124
80	100	142	201	285	403	109	154	218	308	436	121	171	242	342	484	247	355	495	701	991	271	384	543	768	1086
90	97.4	138	195	276	390	105	149	211	298	422	117	165	234	331	469	240	339	480	679	960	263	371	526	743	1052
100	94.5	134	189	268	378	102	144	204	289	409	113	160	227	321	464	232	328	465	658	931	255	360	510	721	1020
120	89.4	126	179	253	358	96.6	134	193	273	387	107	152	215	304	430	219	311	439	622	880	240	340	481	680	962
140	84.8	120	170	240	340	91.8	126	184	260	368	102	144	204	289	409	208	297	418	594	836	228	324	458	648	916
160	81.	115	162	230	325	87.7	123	175	248	352	97.2	138	196	276	390	199	283	399	567	799	218	309	436	619	874
180	77.7	110	156	220	312	83.9	119	168	238	337	93.2	132	187	264	374	191	270	383	541	766	209	296	419	593	839
200	74.7	106	150	212	300	80.8	114	162	229	324	89.6	127	180	254	360	188	260	368	521	737	201	284	403	570	807
225	71.4	101	143	203	287	77.1	109	155	219	310	85.7	121	172	243	344	175	248	352	498	704	191	272	385	546	772
250	69.3	97.1	139	196	279	74.0	105	148	210	297	82.1	116	165	234	330	168	238	338	478	677	184	260	370	522	741
275	66.3	93.8	132	187	265	71.4	101	143	202	288	79.3	112	159	225	318	161	229	324	460	650	177	250	356	502	714
300	63.6	90.0	128	180	256	68.6	96.8	138	195	277	76.2	108	153	217	307	156	221	314	444	629	171	243	344	487	689
350	59.7	84.5	120	169	240	64.2	91.3	129	183	259	71.6	101	144	203	288	146	208	294	417	590	161	228	323	457	647
400	56.1	79.7	113	160	227	60.6	86.1	122	173	245	67.8	95.6	135	192	272	138	196	278	394	557	151	215	304	431	611
450	53.5	75.6	107	152	216	57.4	81.7	115	164	232	63.8	90.3	128	183	258	131	186	264	373	529	143	204	289	410	580
500	50.6	72.	102	145	205	54.7	77.9	110	157	222	61.	86.5	123	174	247	124	177	251	357	505	137	194	276	391	554
600	46.4	66.1	94.1	133	189	50.	71.4	101	144	204	55.7	79.4	113	160	227	114	163	231	329	465	126	178	253	360	510
700	43.	61.3	87.5	124	176	46.4	66.3	94.5	134	191	51.6	73.7	104	149	211	106	151	215	306	433	116	166	236	335	475
800	40.1	57.4	82.	116	166	43.2	62.	88.4	126	178	48.2	69.	98.4	140	198	99.3	142	202	287	407	109	155	221	314	446
900	37.7	54.	77.3	110	156	40.7	57.9	83.5	119	169	45.3	65.	92.8	132	188	93.4	133	190	271	385	102	146	208	297	421
1000	35.6	51.2	73.4	104	149	38.4	55.3	78.1	113	159	43.5	61.6	89.5	126	181	88.2	128	181	261	366	96.9	139	198	282	401
1500	27.9	40.7	59.1	84.7	121	30.2	44.1	63.8	91.6	131	33.7	49.2	71.1	102	146	72.6	102	151	210	299	76.	111	160	230	328
2000	22.7	33.9	48.9	72.2	104	24.7	36.7	54.	78.1	112	27.6	41.1	60.1	87.1	126	58.	85.6	124	180	257	63.8	94.	136	197	284
2500	19.	29.	43.8	63.4	92.1	20.5	31.4	46.7	68.6	99.2	23.	35.	52.2	76.2	110	49.2	73.9	109	158	228	54.1	81.2	119	173	250
3000	15.8	25.	37.7	56.5	82.6	17.2	24.6	41.2	56.1	89.3	18.9	30.4	46.	68.2	99.4	42.	64.6	96.6	142	206	46.3	71.1	106	156	226
4000	12.2	18.2	29	46.1	66.7	11.7	20.3	32.5	49.8	74.2	13.4	25.1	36.6	55.9	83.	30.6	50.2	78.1	117	183	33.9	55.5	85.9	129	189

THE CARRYING CAPACITY OF STEAM PIPES.

THE FLOW OF STEAM IN POUNDS PER MINUTE

TABLE NO 16

INTERNAL DIA. 12" — $C_1 = 2.5$ — 12" STANDARD PIPE. — $L_1 = 88'$ — EXTERNAL DIA. 12.75

L	B.P. 2.3 lbs D=.04352					B.P. 5.3 lbs D=.0507					B.P. 10.3 lbs D=.06253					B.P. 100.3 lbs D=.2617					B.P. 125.3 lbs D=.3147				
	DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE					DROP IN PRESSURE				
	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.	1 OZ.	2 OZ.	4 OZ.	8 OZ.	16 OZ.
10	200	284	401	568	803	216	306	433	613	867	240	340	481	680	962	492	695	985	1392	1970	539	763	1079	1526	2158
20	191	270	382	540	764	206	292	413	584	826	229	323	458	647	916	468	663	936	1327	1876	514	727	1028	1454	2057
30	183	258	366	517	732	197	279	395	558	790	218	309	436	619	876	448	634	897	1269	1795	491	695	982	1391	1965
40	175	248	351	496	702	189	268	379	536	758	210	297	421	595	842	430	609	861	1218	1723	472	667	944	1335	1888
50	169	239	338	478	676	182	258	365	516	730	202	286	405	572	810	415	587	830	1174	1660	454	643	909	1286	1818
60	163	231	326	461	652	176	249	352	499	705	195	276	391	553	782	400	566	801	1133	1602	438	621	877	1242	1755
70	158	223	316	447	632	170	241	341	482	682	189	267	378	535	754	387	548	775	1096	1551	425	601	850	1202	1700
80	153	216	306	433	612	165	234	331	468	662	183	259	367	519	734	376	531	752	1063	1504	411	582	823	1165	1647
90	148	210	297	421	595	160	227	321	454	643	178	252	356	504	713	365	517	731	1034	1462	400	566	800	1132	1601
100	144	204	289	409	579	156	221	313	442	626	173	245	347	491	694	355	502	711	1005	1422	389	551	779	1102	1558
120	137	194	275	389	550	148	210	297	420	595	165	233	330	466	660	338	478	676	957	1352	370	523	741	1047	1482
140	131	186	263	372	526	142	201	284	402	568	157	223	315	446	630	327	456	645	912	1291	358	500	707	1000	1415
160	126	178	252	357	504	136	197	272	385	545	151	213	302	427	604	309	437	619	875	1238	338	479	677	958	1355
180	121	171	242	343	485	130	185	261	370	524	145	205	290	411	581	297	420	595	841	1190	325	460	651	922	1304
200	117	165	234	331	468	126	178	253	357	505	140	198	280	396	561	286	405	574	811	1148	314	445	629	890	1258
225	112	158	224	317	449	120	170	242	342	484	134	190	268	380	537	274	389	550	779	1101	301	426	602	853	1206
250	107	152	215	305	432	116	164	233	329	466	129	182	258	365	517	264	374	529	750	1060	289	409	580	820	1161
275	103	147	208	294	416	112	158	224	318	450	124	175	249	350	499	254	360	510	722	1022	279	395	559	792	1120
300	100	142	201	284	403	108	153	217	307	435	120	170	241	341	483	246	348	493	698	988	270	382	541	766	1083
350	94.1	133	189	266	376	101	144	204	289	409	113	160	226	321	454	231	328	464	657	929	254	359	509	720	1020
400	89.1	126	179	253	359	96.2	136	193	274	389	106	151	214	303	430	219	310	439	622	881	240	340	482	682	966
450	84.7	120	170	241	342	91.5	129	184	260	369	101	144	204	289	409	208	295	418	592	838	228	323	458	649	918
500	80.9	115	163	230	326	87.4	124	176	249	353	97	137	195	276	391	199	282	400	566	802	218	309	438	621	879
600	74.5	106	150	213	302	80.5	114	162	230	326	91.7	127	185	255	362	183	260	369	523	740	201	285	405	573	812
700	69.2	98.5	140	198	281	74.8	106	151	214	304	83.	118	167	238	337	170	242	344	488	691	187	266	377	535	758
800	64.9	92.5	131	187	265	70.2	100	142	201	286	77.8	111	157	224	317	160	228	324	459	651	176	250	355	503	713
900	61.3	87.4	124	176	250	66.2	94.5	134	191	271	74.3	105	151	212	304	151	215	307	434	617	168	236	336	477	676
1000	58.1	83.	118	168	238	62.8	89.8	128	182	258	69.7	100	142	202	286	141	205	286	414	577	158	225	320	454	644
1500	46.6	67.2	96.5	138	196	50.4	72.7	104	148	212	56.	80.7	115	165	235	116	171	238	339	483	130	183	267	378	540
2000	39.2	57.2	82.7	118	169	42.4	61.8	89.3	128	183	47.2	68.7	99.3	142	203	98.4	142	205	299	418	108	157	225	322	459
2500	33.7	49.9	72.8	105	151	36.5	54.	78.7	113	163	40.6	60.	87.4	136	181	85.5	125	181	260	373	94.	137	199	286	409
3000	29.4	44.2	65.1	94.7	146	31.8	47.8	70.4	102	147	35.4	53.2	78.2	114	164	75.4	112	163	236	338	82.9	124	179	261	371
4000	22.5	35.4	53.6	79.8	116	24.5	36.4	58.1	85.9	125	27.4	42.9	64.6	95.6	130	59.8	91.4	136	199	289	66.	101	150	219	317

Synopsis of the paper

The paper consists of a short dissertation on the properties of steam.

A discussion of the different authorities and data from which the capacity of steam pipes are developed.

Gives some reasons why his formula differs from that of Unwin and Babcock. Gives tables showing the capacities of iron pipes of different sizes and lengths for drops of pressure of one to sixteen ounces, and at five different pressures as follows: 2.3 lbs., 5.3 lbs., 10.3 lbs., 100.3 lbs. and 125.3 lbs.

The tables are intended for ready reference for the engineer to enable him to determine the capacity of any size or length of pipe in pounds of steam per hour.

DISCUSSION

Professor Kent: It is to be regretted that Mr. Otis has spent such a great amount of time upon his paper and presented it in such a shape that it needs to be edited and reconstructed before it can be made worthy of a permanent place in our Transactions.

The paper could be improved by the omission of the first six pages, most of which has no relation to the subject of his paper and all of which is only a restatement of facts or theories of elementary physics which may be found (usually in much better form) in text-books on physics or on the steam engine.

A number of pages are devoted to a mathematical discussion of the development of the well-known formula of Unwin as modified by Babcock. A far more satisfactory treatment, occupying only one and one-half pages, will be found in the Babcock & Wilcox Co.'s book "Steam," 1913 edition, page 317.

Babcock's formula is $W = 87 \sqrt{\dfrac{pD\, d^5}{\left(1 + \dfrac{3.6}{d}\right) L}}$ pounds per minute

Otis's formula when reduced to the same form is

$$W = 84 \sqrt{\frac{pD\, d^5}{\left(1 + \dfrac{c_1}{d}\right) L}}$$

in which p is drop in pressure in pounds per square inch, D the density in pounds per cubic foot, d the internal diameter of the

pipe, L the length in feet, and C, in Otis's formula, is a constant $= 3.6$ for a 1 in. pipe, 3.2 for 4 in., 2.95 for 8 in., and 2.6 for 12 in.

Both of the formulae can be reduced to a simpler form, viz.:

$$V = C\sqrt{\frac{p\ D\ d^5}{L}}$$

in which C is a coefficient which varies with the diameter. Using this form we can compare directly the two formulae as follows:

Nominal size of pipe.	1 in.	4 in.	8 in.	12 in.
Value of C Babcock	41.3	63.2	72.3	76.3
Otis	39.4	62.7	71.8	76.2

The difference in the values of C is very small, except in the case of the 1 in. pipe, and in that it is less than 5 per cent. As both the Babcock and the Otis formulae are based on certain assumptions, which may have a far greater error than 5 per cent. for a 1 in. pipe, and as we have no records of actual experiments on 1 in. pipe of different degrees of roughness, we have no means of judging which of the two formulae is more nearly correct.

Mr. Otis gives his formula in the form $W = 84d^3\sqrt{\frac{D\ (p - p)}{L\ (d + c)}}$

and says in italics *The formula upon which the tables are based* They may be based upon the formula, but they are not calculated from it, and the formula from which they were calculated is not given, so that we have no means of checking the figures.

I have made a calculation of the flow of steam, according to Babcock's formula, with a drop of 1 lb. in 200 ft. length, and compared the results with the figures given in Mr. Otis's tables as below:

Normal size of pipe	1	4	8	12
	Flow in pounds per minute.			
Initial pressure $p = 2.3$ lbs.	0.55	28.4	169	468
	(0.66)	(29.8)	(189)	(464)
$p = 10.3$ lbs.	0.72	34.0	203	562
	(0.79)	(36.0)	(226)	(666)
$p = 100.3$ lbs.	1.37	70.0	416	1149
	(1.61)	(73.0)	(464)	(1364)

The figures in parentheses are calculated by Babcock's formula, the others are from Mr. Otis's tables. The differences are quite

erratic, ranging from almost nothing in the case of a 12 in. pipe with p = 2.3 lbs. to nearly 20% in the case of a 1-in. pipe with p = 2.3 lbs. and a 12-in. pipe with p = 100.3 lbs. There is nothing in the paper to explain these differences.

Mr. Otis further says, "These results are practically the same as those obtained by Unwin and Babcock had they made allowance from the variation of flow due to size of pipe, which is not a negligible quantity." He probably means the variation of flow from that calculated simply from the square root of the fifth power of the diameter, but they did make such allowance by putting the expression $1 + \frac{3.6}{d}$ in the denominator. Mr. Otis's reason for modifying the figure 3.6, changing it to C, with values ranging from 3.6 to 2.6 is not clearly explained. His statement that Darcy shows that the value of C varies 8 points in pipes ranging from 2 to 12 inches needs explanation to make it understandable.

It appears that Mr. Otis has modified his formula in some way so as to take in consideration "the frictional loss due to the steam entering the pipe from the nozzle of the boiler," and "the loss of heat from exposure," but he gives no statement, as how these affect the flow in the pipe, nor as to what modifications he has made in the formula on account of them.

He gives us no means by which we may verify the modifications, and no formula by which we may check his table if the modifications are verified. His table therefore must be regarded as having no clearly explained basis.

An inspection of the table shows that the figures vary, very nearly (sometimes exactly) as the square root of the drop in pressure. This corresponds to all the usual formulae and is to be expected. The usual formulae, however, make the flow vary inversely as the square root of the length, but Mr. Otis's figures show a large and unexplained variation from this principle. For example, take 4, 8 and 12 in. pipes, 50, 200, and 800 ft. long, 1 lb. drop, initial pressure 2.3 lbs.

	L =	50		200		800
Size	4	50	(55.6)	28.4	(27.8)	13.9
	8	267	(364)	169	(182)	91
	12	677	(1060)	468	(530)	265

The figures in parenthesis are calculated on the assumptions that the figures for the length 800 ft. are correct, and that the flow for 200 ft. length are just double, and those for a 50 ft. length

just four times the figures for 800 ft. length. Assuming that the figures for 800 ft. length are correct then the computed figures for the 50 ft. length, on the theory that the flow varies inversely as the square root of the length, are respectively 11 per cent., 36 per cent. and 57 per cent. greater than Mr. Otis's figures. These great and erratic differences throw serious doubt on his modified and unrevealed formula, whatever it may be.

Mr. Otis: The Babcock formula is all right so far as it goes, but it assumes that the frictional loss is a constant. This has been demonstrated to be erroneous. Actual experiment shows that the friction due to size is a *variable quantity.* This variable which I have introduced as c_{11} is the only material variation I have made in the formula presented by him.

The formula I have presented is mathematically simplified by eliminating the fifth power of the diameter of the pipe, but this is simply an algebraic calculation, and in no way effects the result. The figure 87 in the Babcock formula is not mathematically correct, nor is the 84 used in my formula exact, although it is nearer than the 87.

I have the variable (c_{11}) with which to correct the error, while the Babcock formula has no correction. If Professor Kent will correct the value of 87 to a number representing the actual sum obtained in the elucidation of his equation, he will find that for a 12 inch pipe the two formulas will almost exactly agree. The variations for the other sizes are due to the variable for frictional resistance.

Professor Kent suggests that both formulae can be reduced to a simpler form, viz.: $V = C\sqrt{\frac{p\,D\,d^5}{L}}$

in which C is the coefficient varying with the diameter. This is mathematically true, as C is not a number, but is the product of two other quantities, and it is questionable as to its simplicity. Also it does not express algebraically the sources of development of the formula. In the formula I have presented c_{11} represents the variable frictional resistance, while C in the above equation would represent only an abstract quantity determined from the value of c_{11} as given. Nothing would be gained, and the quantitive analysis of the formula would be lost.

In order to show that my calculations are wrong he takes as an illustration the flow of steam in 4 in., 8 in. and 12 in. pipe in lengths of 50, 200 and 800 feet, with an indicated pressure of 2.3 lbs. and allowing for a 1 lb. drop at the end of the line.

Since the mathematical calculations for all the sizes of pipe mentioned are similar I will take only one,—that of the 4 in. pipe, in lengths of 50, 200 and 800 feet.

As stated in the paper, frictional resistance from steam entering the pipe from the boiler or a tee is always present and must be taken into consideration, and as this resistance is that of the simple orifice corresponding to the formula

$$L = \frac{114d}{1 + \frac{c_1}{d}} \quad \text{or according to the Babcock formula} \quad \frac{114d}{1 + \frac{3.6}{d}}$$

the value of this frictional resistance for a 4 in. pipe would be 20 feet of its length.

These lengths of resistance then become 70, 220 and 820 feet. These extra lengths are given in the heading of each table, but unfortunately it is not so stated in my paper, which may have misled Professor Kent. It is a fact that my formula would be better expressed as follows:

$$W = 84d^3 \sqrt{\frac{D\ (p - p_1)}{L + L_1\ (d + c_1)}}$$

where the additional factor of L_1 represents a variable of piping to be added for the entrance losses.

If the exposure loss is 7.92 B.t.u. per sq. ft. per degree difference in temperature for 24 hours for standard covering (1 in. thick), the heat loss in a surrounding temperature of 70 degrees for steam at 2.3 lbs. pressure is .00114 lbs. per lineal foot per minute for a 4 in. pipe with a cover 1 1/32 in. thick. (Standard.)

$$\text{Then if } w = 84d^3 \sqrt{\frac{D\ (p - p_1)}{L + L_1\ (d + c_1)}}$$

$$\begin{aligned} \text{and } d &= 4.026 \\ p - p_1 &= 1 \\ D &= .04352 \\ L &= 50 \qquad = 70 \\ L_1 &= 20 \\ c_1 &= 3.2 \end{aligned}$$

$$\text{We have } w = 84\ (4.026)^3 \sqrt{\frac{.04352}{(7.226)\ 70}} = 50.81$$

or if $L = 220$ $w = 28.66$

or if $L = 820$ $w = 14.95$

The heat loss for 50 ft. will be .05 lbs.
The heat loss for 200 ft. will be .23 lbs.
The heat loss for 800 ft. will be .91 lbs.
Deducting these losses from the value of w:
We have for 50 ft. of pipe a delivery of 50 lbs.
We have for 200 ft. of pipe a delivery of 28.43 lbs.
We have for 800 ft. of pipe a delivery of 13.94 lbs.
The value given in the table.

The Babcock formula would make these values as follows:
For a pipe 50 ft. in length 60.63 lbs. per minute.
For a pipe 200 ft. in length 30.31 lbs. per minute.
For a pipe 800 ft. in length 15.15 lbs. per minute.

Thus you will note that for 50 ft. lengths Professor Kent would have a quantity 19% in excess of what can actually be obtained. This variation being mainly due to resistance from steam entering the pipe.

As stated in the paper, I have no data on the resistance of fittings, and this has led to the omission on my part of an important factor in determining the size or carrying capacity of long runs of pipe interspersed with branch tees and size reductions. The Publication Committee, however, desires my opinion on this matter and it is herewith given.

What frictional loss occurs, in a reducing tee when the steam does not change its direction of flow, is problematical. In practice I have estimated this to be about 40 per cent. of the loss in a reducing tee or boiler nozzle.

If this loss be accepted as 40% and the loss given the boiler nozzle has been figured in the length of pipe as given in the table, it is evident that this deduction in the length of pipe as given must be made if there is no change in direction of flow of steam.

As an example, and at the request of the Publication Committee, let us take a line 800 feet in length and reducing in size every 50 feet by a reducing tee leading to a branch corresponding to the reduction in the size of the main as follows:

Boiler 8″	6″	5″	4″	end
200′	200′	200′	200′	
200′ = 84 lbs.	169′ = 45 lbs.	183′ = 27.5 lbs.	188′ = 14.5 lbs	

Boiler pressure 2.3 lbs. Drop at end of line 1 lb.

How much steam will the line carry under a uniform drop in pressure?

As the 8 in. pipe takes steam from the header and admits of a 4 oz. drop, the tables give the carrying capacity of 84 pounds per minute.

As the 6 in. section receives the steam directly from the 8 in line we will assume the entrance resistance as 40% of the header resistance, or 40% of 52 ft. = 21 ft.

As this resistance was taken at 52 ft. in forming the tables it is evident that 31 feet should be taken from the 200 ft., leaving 169 ft.

For this length the tables give for a 6 in. pipe 45 lbs.

For a 5 in. pipe the corrected length would be 183 ft. and the carrying capacity as shown by the tables would be 27.5 lbs.

For a 4 in. pipe the correct length would be 188 ft. and the carrying capacity 14.5 lbs.

Reverse the problem:

What sizes of pipe are required to meet the following conditions?

Total length of main 800 feet. Boiler pressure 2.3 lbs. Drop at end of line not over 8 oz.

5000 lbs.	4000 lbs.	3000 lbs.	2500 lbs.
120′	135′	125′	100′
120′ = 8″	98½′ = 9″	93¾′ = 8″	73.6′ = 7″
2000 lbs.	1200 lbs.	800 lbs.	400 lbs.
80′	90′	80′	60′
58.4′ = 6″	73.2′ = 5″	68′ = 4″	50′ = 3½″

As there are eight divisions we will assume the drop in pressure to be 1 oz. to each section, and as the flow is in pounds per hour we will divide each quantity by 60 to meet the requirements of the table.

5,000 lbs. per hour = 83 1/3 lbs. per min. for 120 feet = 10 in. pipe.

4,000 lbs. per hour = 66 2/3 lbs. per min. for 98½ feet = 9 in. pipe.

3,000 lbs. per hour = 50 lbs. per min. for 93¾ feet = 8 in. pipe.

2,500 lbs. per hour = 41 2/3 lbs. per min. for 73.6 feet = 7 in. pipe.

2,000 lbs. per hour = 33 1/3 lbs. per min. for 58.4 feet = 6 in. pipe.

1,200 lbs. per hour = 20 lbs. per min. for 73.2 feet = 5 in. pipe.

800 lbs. per hour = 13 1/3 lbs. per min. for 68 feet = 4 in. pipe.

400 lbs. per hour = 6 2/3 lbs. per min. for 50 feet = 3½ in. pipe.

The 4 in. and 5 in. sizes will have a drop in pressure of a little more than 1 oz. in the run, but as the steam comes to them with less than 5 oz. drop, the full drop at the end of the line would probably not exceed 8 oz.

If a one pound drop could be allowed the sizes would be 9 in., 8 in., 7 in., 6 in., 6 in., 5 in., 4 in., 3 in.

If a boiler pressure of 5.3 lbs. is maintained and a one pound drop allowed, the pipe sizes would be 9 in., 8 in., 7 in., 6 in., 5 in., 5 in., 4 in., 3 in.

In estimating branch lines, having sub-branches, start the size from the full length as given in the tables, and for future branches make the 40% reduction.

CCCLXXX

RATIONAL METHODS APPLIED TO THE DESIGN OF WARM AIR HEATING SYSTEMS.

BY ROY E. LYND, MEMBER.

The one form of heating which has suffered more than any other through the lack of reliable scientific information pertaining thereto, is warm air heating. A great deal of the work done in the past in connection with warm air heating has been largely of the hit-or-miss variety, and it is astonishing, the number of otherwise well informed people who believe that warm air installations *cannot* be designed according to scientific principles—who believe that there are *no* formulae which will give results which accord with the proportions found in successful installations. It is to contribute something, small though it may be, toward dispelling this illusion, as well as for the purpose of increasing the somewhat scanty collection of data pertaining to this subject which is to be found in the Transactions of this Society, that this paper is presented. The author has endeavored to present in graphical form in Chart 1, practically all the data needed by the designer in proportioning warm air pipes to heat any ordinary room, after the heat losses have been calculated by some reliable rule. In Chart 2 is found a graphical representation of Warm Air Heater capacities under different conditions of efficiency, and also the free air area through the heater under average conditions.

Chart 1 is obtained as follows:—The curves in the upper left-hand corner are plotted with register temperatures as abscissae, and cubic feet of air delivered per hour through a pipe of one square foot cross-sectional area and an effective elevation of thirty-five feet, as ordinates. The ordinates were calculated from the formula—

$$Q = 3600\ C\ A\ \frac{T_1}{T_2}\sqrt{2gh\ \left(\frac{T_1 - T_2}{T_1}\right)}$$

in which Q = cubic feet of air issuing from register per hour.

C = the ratio between the actual velocity and the theoretical velocity.
A = one square foot.
T_1 = absolute temperature of air leaving register.
T_2 = absolute temperature of air entering heater chamber.
g = acceleration due to gravity.
h = effective height of warm air flue.

This formula is the same as Snow's velocity formula, multiplied by 3600 to change from cubic feet per second to cubic feet per hour, and multiplied by the co-efficient of friction of the system. This co-efficient has been taken as 1/3 for first floor rooms, and ¼ for second and third floor rooms. In calculating velocities in a warm air system, the author has found that a value of h equal to 6 feet for first floor rooms, 16 for second floor rooms, and 25 for third floor rooms give results which accord most closely with actual measured velocities, and taken in conjunction with the above friction co-efficients give calculated velocities, quantities of air, and pipe sizes which agree very well with the best practice. These values of h are practically based on the assumption that the lowest point of the effective flue height is the top of the firepot, and the highest point is the floor line of each story. For example, if the height from the cellar floor to the top face of the main floor is eight feet, the effective flue height to the first floor would be six feet, as the top of the firepot is usually about two feet above the cellar floor. If the height of the first story rooms is nine feet and the floor construction one foot thick over all, this would add ten feet to the six, making sixteen feet in all for the effective flue height to second floor rooms. Likewise, we add eight feet for the height of the second floor rooms and one foot for the floor thickness, making a total effective height of twenty-five feet to third floor rooms. This assumption disregards the fact that side-wall registers may be a little above the floor, but it fits actual conditions and actual results better than any other assumption which could be made.

The reason a different co-efficient of friction has to be used for second and third floor rooms than that used for first floor rooms is due, no doubt, to the fact that in the average installation flat wall pipes are used to convey the warm air to second and third floor rooms, whereas round pipes with easy bends are run to first floor rooms. At any rate, these are the actual ratios we find in practice, and in order to be conservative we must reckon with them.

The curves plotted as above are figured on the basis of air entering the heater chamber at zero, 10 degrees, etc., up to 70 degrees, in order to fit all conditions between the two limits of outside air

supply and indoor re-circulation, and are designated 1, or 2 and 3, according as they refer to first, or second and third floors.

In order to apply these curves to any effective flue height, as well as to the assumed thirty-five feet, the sloping lines in the upper righthand corner of the chart were drawn. It can be seen from formula (1) that the quantity of air delivered is proportional to the square root of the height. Therefore, in order to find the value of

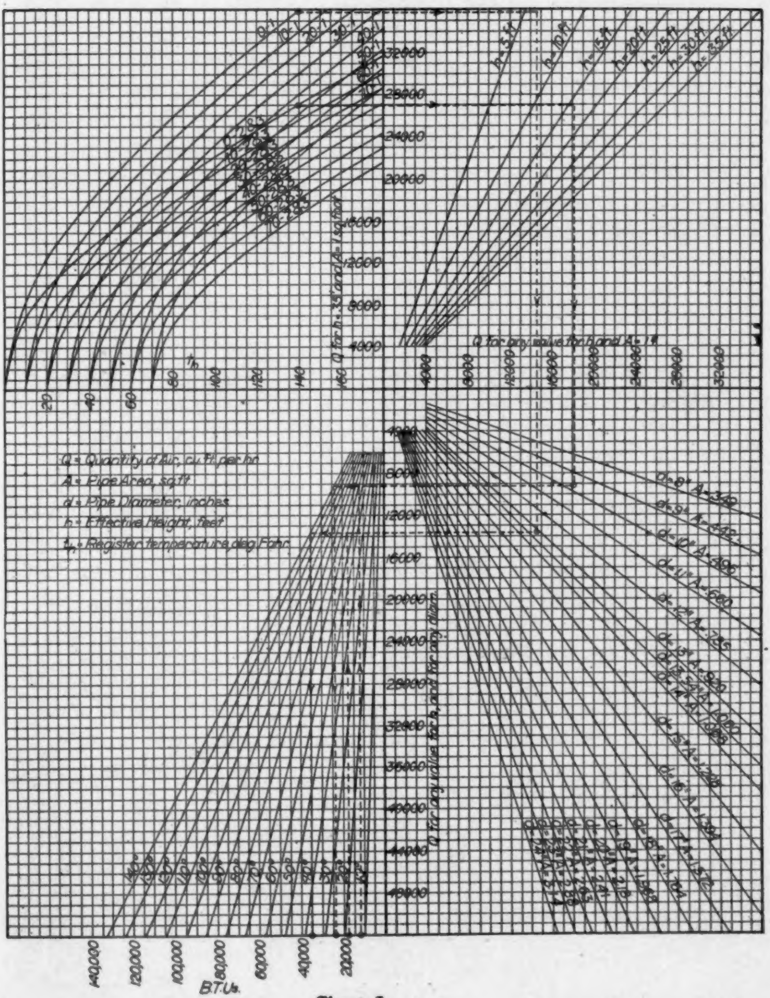


Chart I

Q for an effective flue height of five feet, it will be necessary to multiply the value for thirty-five feet by $\sqrt{\frac{5}{35}}$ or .379. This is a straight line relation, and the line marked h = 5 feet is plotted by making the abscissa of any point equal to .379 times the ordinate. The lines for 10, 15, 20, 25 and 30 feet are plotted in the same manner, the line marked h = 35 feet standing at an angle of 45 degrees, as its ordinate is equal to its abscissa at every point.

This gives us along the right-hand half of the horizontal axis a little above the center of the chart, the value of Q for any height, and for a pipe with a cross-sectional area of one square foot. To get the value of Q for any value of h and a pipe of any cross-sectional area, we use the sloping lines in the lower right-hand corner of the chart. It can be seen from formula (1) that the value of Q is proportional to the value of A, and consequently, to the value of d^2. For an eight-inch pipe, we would have to multiply the value of Q as found above by $\left(\frac{8}{13.54}\right)^2$, 13.54 being the diameter in inches of a pipe with a cross-sectional area of one square foot. This is, again, a straight line relation, and the line for an eight-inch pipe is plotted by making the ordinate of any point equal to $\left(\frac{8}{13.54}\right)^2$ or .349 times its abscissa. The lines for pipes from 8 inches to 24 inches are plotted in the same manner. Nothing smaller than eight inch is given, as the author does not consider it advisable to use pipes smaller than eight inch under any circumstances with a gravity system.

This brings us over to the lower half of the central axis of the chart, with values of Q for any effective height and any cross-sectional pipe area. The sloping lines in the lower left-hand section of the chart express the relation between the cubic feet of air and the heat in B.t.u.'s which the air carries. The formula from which these lines are derived is as follows:—

$$H = \frac{Q\,(t_1 - t_2)}{55} \qquad (2)$$

in which H = the heat in B.t.u.'s.

t_1 = higher temperature of the air.

t_2 = lower temperature of the air.

It can be seen from formula (2) that if a given quantity of air, Q,

is to be heated, or cooled, through a given range of temperature, as $t_1 - t_2 = 70$ degrees, it is simply necessary to multiply Q by $\frac{70}{55}$ or 1.273 in order to find the amount of heat added to or extracted from the air. The line marked 70 degrees has therefore been plotted by making its abscissa 1.273 times as long as its ordinate. The other lines, ranging from 10 degrees to 140 degrees have been plotted in the way. It will be noted that in this last group of lines it has been found necessary to change the scale of abscissae, in order to get the chart within a reasonable compass, but this does not affect either the logic or the accuracy of the work.

The manner of using this chart, together with the manner of using Chart 2, will be explained after briefly describing Chart 2. Chart 2 shows graphically the total heat transmitted to the air passing through a heater chamber of any grate diameter per hour, and also the grate area and free air area of chamber of any size heater. The curves in the left-hand half of the chart are plotted by making the ordinate of any point equal to the diameter of the grate in inches, and its abscissa equal to the total heat transmitted to the air per hour, when the rate of combustion over the actual grate area is 6 lbs. per square foot of grate per hour, and the heat utilized, or the product of efficiency $\times$ calorific value of the fuel is equal to 6500, 7000, 7500, 8000, 8500, and 9000 B.t.u.'s, for each of the curves so designated. These are the values which the author has found to cover about all the types of warm air heater construction on the market to-day, ranging from 6500 on the cheap, speculative heaters, to 9000 on the most efficient types of heaters. The calculation of the length of the abscissa of any point on any curve, would be as follows:—

$$H = G \times R \times E \times F \qquad (3)$$

in which G = grate area in square feet.

R = rate of combustion in pounds per square foot of grate per hour.

E = efficiency of heater, per cent.

F = calorific value of the fuel in B.t.u.'s per pound.

Now, familiarity with different types of heaters enables us to place a given heater in the 7500 B.t.u. class, for example, or in the 8000 B.t.u. class, etc., when we know something about its construction, area and arrangement of heating surface, etc., so that to apply formula (3) to a furnace in the 8000 B.t.u. class, for example, at a 6 pound rate of combustion, we make $R = 6$, $E \times F = 8000$, and the formula becomes,

$$H = 48000\,G$$

from which we can easily plot the 8000 curve. The others are plotted in the same way.

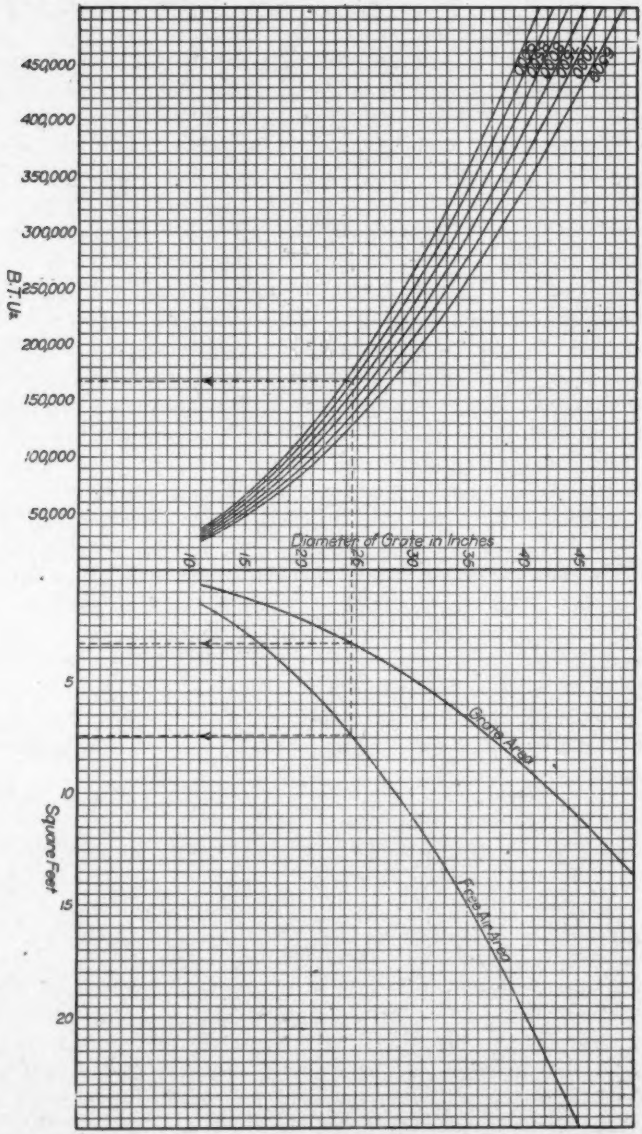


Chart II

The curves to the right of the center line represent the grate area in square feet and the free air area in square feet for any grate diameter. The grate areas, of course, are figured from the grate diameter in the usual way. The free air area is obtained by multiplying the grate area by 2.25, the author having found from a large number of measurements of different kinds and types of heaters, that the cross-sectional area of the passages through the heater chamber by which the heat of the furnace is imparted to the air will average 2.25 square feet per square foot of furnace grate area. This is an average, and in any particular case it would be better to know what the actual free air area is, but this information is sometimes difficult to obtain from the manufacturers, and the author does not remember ever having seen it in a catalog.

The method of using these charts can best be explained by working out a specific example. Three plans are shown, representing the cellar, first floor, and second floor of a house.

The measurements of the rooms, glass and wall surface, etc., together with the calculated heat losses are given in Table 1. The heat losses are figured by Carpenter's rule.

$$H = 70\ (G + \tfrac{1}{4}\ W + .02n\ C) \qquad (4)$$

in which H = heat losses in B.t.u.'s per hour.

G = exposed glass surface in square feet.

W = net exposed wall surface in square feet.

n = number of changes of air per hour.

C = contents of room in cubic feet.

In the application of this rule to the example given, n has been taken as 1.

TABLE 1

Rooms	Wide	Long	High	Cubic Feet	Exp. Glass	Net Exp. Wall	Heat Losses	Exp. Allowances	Total Heat Losses
FIRST FLOOR									
Living Room	14	21	9	2646	94	302	15610	+1561	17171
Den	10	12	9	1080	44	172	7630	+763	8393
Dining Room	12	16	9	1728	44	100	7280		7280
Pantry	6	17	9	918	15	57	3290		3290
Hall (front)	6	18	9						
Hall (rear)	6½	14½	9	3699	68	322	15610	+1561	17171
Hall (sec. floor)...	6½	34	8½						
SECOND FLOOR									
Cham. No. 1........	14	17	8½	2023	33	231	9170		
Dressing Room ...	6	8	8½	408	15	36	2240		11410
Cham. No. 2........	12	15	8½	1530	30	200	7770	—777	6993
Cham. No. 3........	12½	16	8½	1700	30	327	10220	—1022	9198
Bath Room	8	9	8½	612	12	56	2600		2600
				16344					83506

Allowances for exposure have been made in Table 1 by adding 10 per cent. to north or west rooms on first floor, and to halls, and deducting 10 per cent. from south rooms on second floor except bath rooms.

The plans show an actual heating installation which has been found to give splendid satisfaction, and it is the purpose of the author to compare the proportions found in that installation with the proportions which would be found by using Charts 1 and 2. The actual system is designed for outdoor cold air supply entirely. The charts have been used to calculate such a system; and also one for the same house providing ventilation for six occupants, or two to each chamber, and recirculating within the building such excess air as may be necessary as a heat carrier; and also one where all of the air is recirculated, as warm air systems are usually installed through the middle west. The results of these operations are given in Table 2.

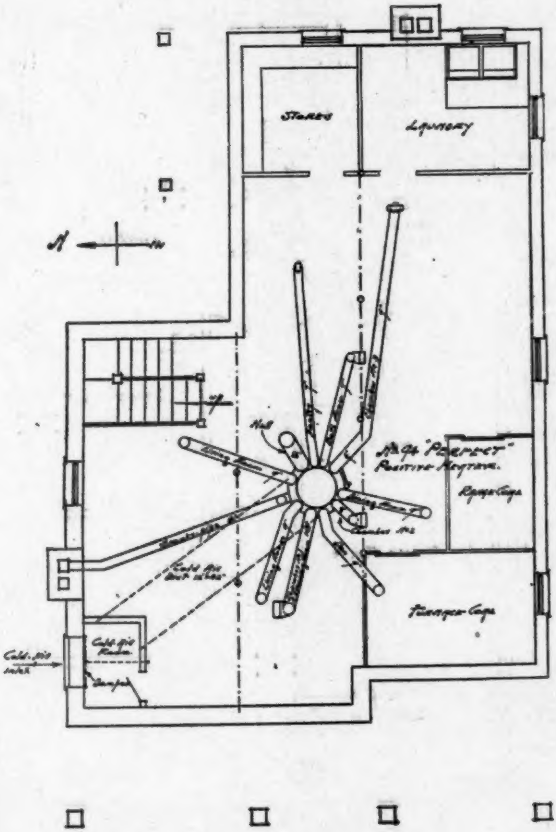
TABLE 2

	Actually Used		Outdoor Air Supply From Chart 1		Ventilation for 6 persons From Chart 1		Recirculated Air Entirely From Chart 1	
Rooms	Diam. Pipe	Area Pipe	Diam. Pipe	Area Pipe	Diam. Pipe	Area Pipe	Diam. Pipe	Area Pipe
FIRST FLOOR								
Living Room ... }	9	.442	9	.442	11	.660	12	.785
	9	.442	10	.496	12	.785	12	.785
Den	9	.442	9	.442	12	.785	12	.785
Dining Room ...	9	.442	9	.442	10	.496	11	.660
Pantry	7	.267	8	.349	8	.349	8	.349
Hall	12	.785	12	.785	15	1.228	15	1.228
SECOND FLOOR								
Cham. No. 1... } Dressing Room. }	10	.496	10	.496	12	.785	12	.785
Cham. No. 2....	8	.349	8	.349	9	.442	10	.496
Cham. No. 3....	9	.442	9	.442	10	.496	11	.660
Bath Room	7	.267	8	.349	8	.349	8	.349
		4.374		4.592		6.375		6.882

In Table 2, the pipes actually used were copied from the actual plan. The method of arriving at the values for outdoor air supply from Chart 1 is as follows:—

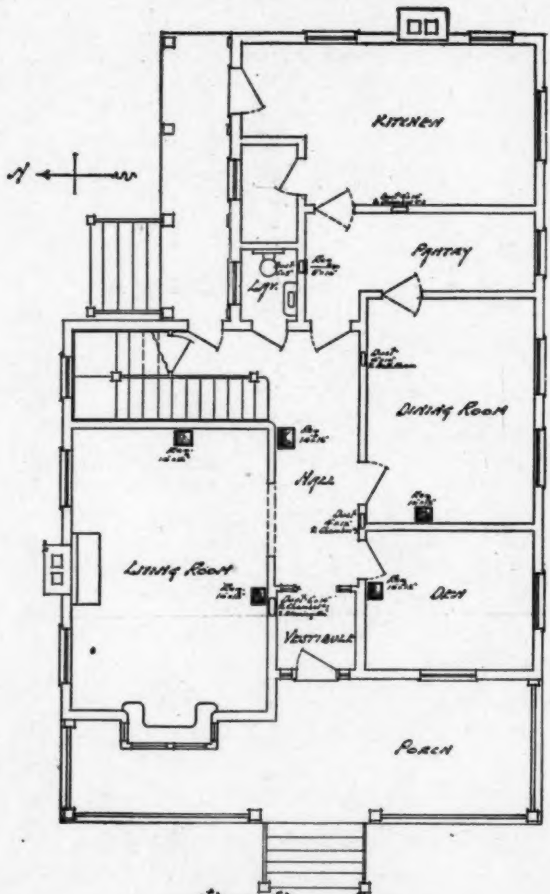
On the left-hand side of the bottom line of the chart find a point representing the heat loss from the living room on the first floor, or 17171 B.t.u.'s. Project this point up until it meets the sloping line marked 70 degrees, as the air is to cool through 70 degrees, or from 140 degrees to 70 degrees, in heating the room. Project the intersection horizontally into the right-hand half of the chart. Now, as we are assuming that the air enters the heater at zero, and leaves the register at 140 degrees, and we are considering a first floor room, find a point in the upper left-hand corner of the chart where the vertical line representing 140 degrees intersects the curve marked 0—1. Project this intersection horizontally into the right-hand half of the

House Plan 1



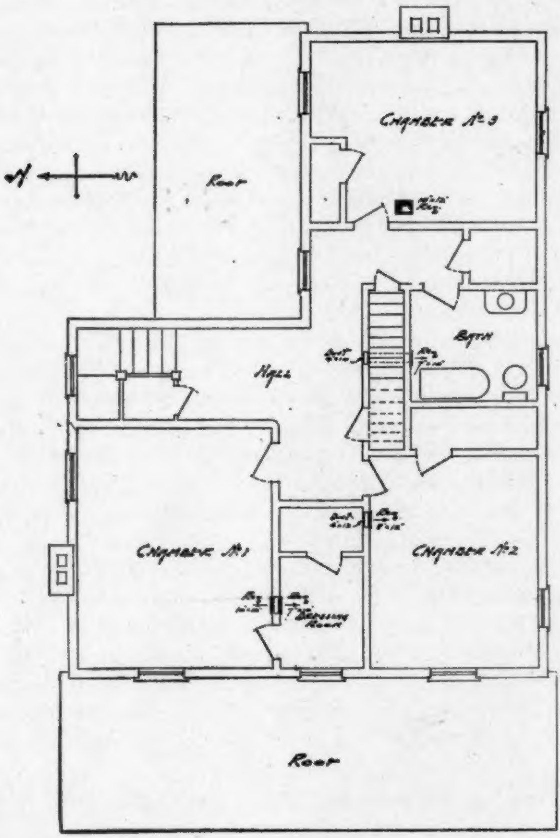
Cellar Plan

House Plan 2



First Floor Plan
Scale 1/8"=1'0"

House Plan 3



Second Floor Plan
Scale 1/8"=1'0"

chart until it intersects an oblique line representing a six-foot effective flue height. Project this intersection downward until it intersects the horizontal line first projected into this section of the chart. This last intersection is nearer the 13-inch pipe line than any other, so a 13-inch pipe would have to be used. Instead of this, two pipes were actually used on account of the length of the room, and to equal a 13-inch pipe in area, they would have to be one 9-inch and one 10-inch pipe. This whole operation is illustrated by one of the dotted lines on Chart 1, the other one representing a similar calculation for Chamber No. 1 on the second floor. In this latter case the operation was the same, except that the curve 0—2 and 3 was used instead of 0—1. It might also be mentioned that as only one register was used in the hall, and the hall was two stories in height, the mean between curves 0—1 and 0—2 and 3 was used, as was also an average effective flue height of $\frac{6 + 16}{2}$ or 11 feet. The seven-inch pipes used in the actual installation to heat the pantry and the bath-room were amply large to heat those rooms, as far as cross-sectional area was concerned, but the author believes it good practice to use nothing smaller than eight-inch pipes in any case.

Eliminating the two 7-inch pipes, the only difference between the actual installation and our computed pipe sizes is the fact that we specify a 9-inch and a 10-inch pipe to heat the living room, whereas two 9-inch pipes were actually used.

The above data all refers to the condition that all of the air is to be taken from outside. This is the general practice in this part of the country, but as a matter of fact, it provides far more ventilation than is considered necessary with any other kind of a system. Referring to Table 1, the total heat losses are seen to be 83566 B.t.u.'s per hour. Now as one cubic foot of air in cooling from 140 degrees to 70 degrees, gives up 1.1 B.t.u.'s, the number of cubic feet of air necessary as a heat carrier would be $\frac{83566}{1.1}$ or 75969 cubic feet per hour. The total cubic contents of the house from Table 1, is 16344 cubic feet, so if all the air is taken from outdoors, we have $\frac{75969}{16344}$ or 4.65 complete changes of air per hour; or we have ventilation for $\frac{75969}{1800}$ or 47 persons. It is not likely that more than six persons

would occupy the house, so that 6 by 1800 equal 10800 cubic feet of air per hour is all the air that is actually needed for ventilation. The one change of air per hour which we have figured in our heat losses would, therefore, be more than ample for ventilation, but leakage is a rather uncertain quantity. It, therefore, seemed wise to the author to figure this installation out three ways: 1st, with all the air taken from outside; 2nd, with only enough air taken from outdoors to provide positive ventilation for six persons; and, 3rd, with all the air recirculated, allowing for the air leakage taking care of the ventilation requirements.

The method of determining the pipe sizes for the first case has been outlined above. The method for each of the other cases is the same, except that we must take into consideration the temperature at which the air enters the heater in each case. With total recirculation, this would be 70 degrees, and we use the curve 70—1 for first floor rooms, and 70—2 and 3 for second and third floor rooms. To determine the average temperature of the air entering the heater in the second case, where ventilation is provided for six persons, we have recourse to formula (2). The method is to determine how much heat the air from the house and the air from outside, each carry into the heater above 0 degrees, and then find at what average temperature the whole quantity of air would have to enter the heater to carry the same amount of heat. The application of formula (2) to this case is as follows:—

$$H = \frac{Q\,(t_1 - t_2)}{55} \qquad (2)$$

$$H = \frac{10800 \times 0}{55} = \frac{(75969 - 10800)\,70}{55}$$

$$= \frac{65169 \times 70}{55}$$

$= 82920$ B.t.u.'s entering heater per hour, above 0 degrees.

Now if t_e = average temperature of entering air,

$$82920 = \frac{75969\,(t_e - t_2)}{55}$$

$$\text{or } t_e = \frac{82920 \times 55}{75969}$$

$= 60$ degrees.

We, therefore, use for the second case the curve 60—1 for the first floor, and 60—2 and 3 for the second and third floor, in determining the required pipe sizes.

Now as to the size of the air inlets to the heater. It is considered good practice, with outdoor air supply to make the cross-sectional area of the air inlet equal to .8 of the total area of all the pipes. It therefore seems logical to make the area of the air inlet, with indoor recirculation, .9 of the total area of the pipes. It would also seem logical, in the case of a combination of the two as in our second case, to divide the total pipe area in the same proportion as the two quantities of air, one from outside and the other from inside the building, and then make the inlet from outside .8 of its proportion, and the inlet from indoors .9 of its proportion. Using the pipe areas as determined in Table 2, we have for the three cases:—

Case 1. C. A. Inlet $= .8 \times 4.592$

$= 3.67$ square feet.

Case 2. C. A. Inlet $= \dfrac{.8 \times 10800 \times 6.375}{75969}$

$= .725$ square feet.

W. A. Inlet $= \dfrac{.9 \times 65169 \times 6.375}{75969}$

$= 4.925$ square feet.

Case 3. W. A. Inlet $= .9 \times 6.882$

$= 6.19$ square feet.

The C. A. inlet under case 1 agrees very well with the C. A. inlet actually used, which, as shown on the plans, was 12 inches by 42 inches, and had a cross-sectional area of 3.5 square feet.

We now come to the final step in the analysis of this problem—the selection of a heater. It was decided in this case to use a Perfect Positive Heater No. 24 with a 24-inch grate. This type of heater will, as the author knows from personally conducted tests, transmit to the air passing through its heating chamber under normal conditions about 8500 B.t.u.'s per pound of ordinary anthracite coal. For case 1, we therefore take the total number of B.t.u.'s which the heater must deliver, equal to 2×83566 or 167132 B.t.u.'s per hour, step this off to the left of the center on the lower axis of Chart 2, project this up until it intersects the 8500 curve, project this to the right horizontally, and where it intersects the center vertical axis we find that the heater must have a grate diameter of 24.6 inches; by continuing the horizontal line until it intersects the two curves to the right of the center, we find that the grate area of this heater will be

3.3 square feet, and the free air area 7.4 square feet. This free air area is more than ample, as the sum of the pipe areas is only 4.592 square feet. We could not, however, use a much smaller heater without requiring an excessive rate of combustion. This operation is shown by the dotted lines on Chart 2.

When we come to case 2 and case 3, however, we have to reverse the operation, as a heater selected on the basis of heat requirements and a 6 pound rate of combustion would not have a sufficient free air area to handle the air which must pass through it. The area of all the pipes in case 2 is 6.375 square feet. A heater with this amount of free air area would have a grate 22.8 inches in diameter, and to deliver to the air 98630 B.t.u.'s would necessitate a rate of combustion of 4.1 pounds per square foot per hour. Likewise in case 3, the heater would have to have a free air area of 6.882 square feet, a grate diameter of 23.8 inches, and a rate of combustion of 3.2 pounds per square foot of grate per hour.

Now the No. 94 Perfect Positive Heater, which was actually used in this installation, comes the nearest to fulfilling the requirements of all three cases. Its grate diameter is 24 inches, and its free air area (from the chart) is 7.06 square feet. The grate area is about right for case 1, and would necessitate a maximum rate of combustion of 6.3 pounds per square foot of grate per hour in that case; and its free air area is about right for cases 2 and 3, and in the two latter cases the maximum rates of combustion would be 3.7 and 3.1 pounds respectively.

All of the above demonstrates a fact which the author has always maintained—viz. that it takes practically the same size heater for a given building, whether the air is taken all from the outside, or all from the inside, or part from outside and part from inside.

Synopsis of the paper

Warm air heating has suffered for lack of engineering data, and this paper supplies the want to a considerable extent.

Two charts are supplied, one relating to pipe sizes and the other to furnace capacities, and the data from which they are drawn is fully given.

A plan of a typical residence is illustrated and the data as given in the paper is applied to the heating plant of the residence shown.

The value of recirculating air is also considered.

DISCUSSION

Mr. D. M. Quay: I would like to ask the author of the paper a question as to the recirculation of the air. Do you recirculate from the main room or recirculate from each room?

Mr. R. E. Lynd: As I understand it the general practice is to recirculate from a large register in the main hall in front of the door. I suppose it would be practical to do it the other way, but I don't know of any installations where that has been done.

Mr. D. M. Quay: I am sure that we can accomplish very much more satisfactory work by recirculating from each of the rooms, and I am very strongly of the opinion that furnace heating could be made as nearly perfect as any other kind of heating by taking return flues from the outside of the room where you bring the air in, and take the room air back to the furnace, one can heat the entire room under almost all conditions, which cannot be done without some means of displacing the air in the rooms.

This is not the first time I have spoken of this problem in our transactions and I have been hoping that some of the furnace heating engineers would bring in a paper or some information showing some results of this method of furnace heating. I am just as positive of the improvement of furnace heating by this method as with the improvement that has been made on any other class of heating.

Prof. Kent: Chart 1 is correct theoretically. If we have a vertical pipe leading to a room and keep the air inside warm to a certain temperature, we may calculate how much hot air would flow up, provided both the entrance and the exit are unobstructed, but when we have a pipe discharging into a room which has no outlet and the amount of air that can enter that pipe is also limited, the formula fails. In order to get warm air to flow into a room the pressure in the room must be lower than the pressure in the chamber from which the air is supplied to the pipe, and air cannot flow into a room unless there is a provision for it to flow out. The remedy, of course, as Mr. Quay says, is that in future every room will have an outlet, so that the air will be recirculated.

The statement of the paper that it "takes practically the same sized heater for a given building, whether the air is taken all from the outside or all from the inside, or part from outside and part from inside," is true on account of the fact that we can get more heat out of a heater than the normal amount provided we

burn enough coal. If the air is taken from the outside the amount of coal required will depend on the outside temperature.

Mr. R. E. Lynd: I have been up against the same difficulties that Prof. Kent speaks of and this data is the nearest I have been able to evolve as to a rational method of solving the problem. I have measured the velocity of air entering the closed room from register with no outlet from the room other than its natural outlets. I find the co-efficients I have used correct in a majority of cases. It is much more accurate than the ordinary method of designing hot air furnaces. We may from this develop some method which will be correct.

Mr. E. S. Mobley: I find in the practice of hot air heating it is very desirable to get an outlet from all rooms back to the furnace chamber and then there is no trouble in getting hot air to the rooms. I invariably put return circulating registers in the rooms of all residences I heat.

Mr. W. T. Colbert: It is pleasing to note the awakening of interest in the possibilities of warm air furnace heating. The public will be much benefited by the preparation and wide dissemination of reliable information on this subject.

The form of the paper presented by Mr. Lynd shows that he has spent much time in preparation of the material, which, unfortunately, is of little value because it is based on false hypotheses.

The velocity of flow of air through piping in a warm air heating system does not depend on the formula

$$Q = 3600\,C\,A\,\frac{T_1}{T_2}\,.\sqrt{2gh\left(\frac{T_1 - T_2}{T_1}\right)}$$

when the air supply to the furnace is drawn directly from outdoors and the exhaust from the rooms is dependent on the accident of leakage, unless the leakage from the rooms is equal to or greater than the volume of air required to heat the room at 140 deg. register temperature.

In practice, I have yet to see a warm air heating system depending on outdoor air supply to furnace and natural leakage from rooms that would operate at a register temperature of less than 180 deg. in zero weather and in exposed buildings the required register temperature has proved to be higher—a condition that is altered but little, when larger heat-pipes are substituted.

This method of furnace installation should be discarded, because theoretical proportioning of heat-pipes cannot take care

of the accidents of too tight or too loose building construction, pressure of prevailing winds, etc., which are the important factors in determining the success or failure of the outside air supply and leakage exhaust type of installation.

The formula:

$$Q = 3600\ C\ A \frac{T_1}{T_2} \sqrt{2gh \left(\frac{T_1 - T_2}{T_1}\right)}$$

does apply with reasonable accuracy to return air circulation methods of furnace installation; but for register temperatures of 140 deg. I would recommend the following changes in values.

C becomes .575 for 1st floor, .50 for 2nd floor and .475 for 3rd floor when horizontal length of cellar heat-pipe does not exceed 5 ft.

T_1 becomes average absolute temperature in heat-pipes (instead of absolute register temperature).

T_2 becomes absolute temperature of air at furnace intake (absolute temperature of room minus 10 deg.).

Then our formula gives:

for first floor

$$Q = 3600 \times .575 \times 1 \times \frac{610}{520} \times \sqrt{2 \times 32.2 \times 6 \left(\frac{610 - 520}{610}\right)} =$$

18,331 cu. ft. per hr.

Velocity 305.5 ft. per minute or 5.09 ft. per sec.

for second floor

$$Q = 36 \times .5 \times 1 \times \frac{610}{520} \times \sqrt{2 \times 32.2 \times 16 \times \left(\frac{610 - 520}{610}\right)} =$$

26,041 cu. ft. per hr.

Velocity 434 ft. per minute or 7.23 ft. per sec.

for third floor

$$Q = 36 \times .475 \times 1 \times \frac{610}{520} \times \sqrt{2 \times 32.2 \times 25 \times \frac{610 - 520}{610}} =$$

30,924 cu. ft. per hr.

Velocity 515.2 ft. per minute or 8.59 ft. per sec.

When heat-pipes have more than 5 ft. horizontal length and are tortuous, the size must be increased and special insulation pro-

vided to offset the increased friction and the cooling of air in transit to register.

It is a fact that the velocity of flow of air is increased and register temperature is decreased when the air supply to a furnace is changed from outside air to the recirculating type, without changing the heat-pipes, which is contrary to Mr. Lynd's deduction. I have proved this in my own house.

It makes practically no difference in heat piping or in furnace capacity when air for ventilation for 6 persons is introduced from outside.

In the all-recirculated-air type of installation all the air is not returned from the upper floors and this deficiency is made up by abnormal inward leakage around windows and doors on the 1st floor.

Introducing sufficient air from outdoors to supply a reasonable amount of fresh air for occupants of the building simply reduces inward leakage of cold air into 1st floor rooms.

The formula for heat delivery given by Mr. Lynd is in error because the air is measured at register temperature 140 deg. while the value for heat in 1 cu. ft. of air is based on the heat required to raise the temperature of air measured at 70 deg.

For room temperature 70 deg. and for any register temperature the formula should read

$$H = \frac{Q \dfrac{530^\circ}{t_1} (t_1 - t_2)}{56} \text{ instead of } H = \frac{Q\,(t_1 - t_2)}{55}$$

In which the added factors 530 deg. represents absolute temperature of air in room and t_1 and t_2 represent absolute register temperature and absolute room temperature respectively.

This gives for 140 deg. register temperature
$H = Q\ 1.1$ B.t.u.
making the balance of Mr. Lynd's calculations only 87 per cent. of the proper amount.

This would increase the size of all the heat-pipes cited in his examples, even though they now are abnormally large for recirculated air.

Taking the house in the example cited by Mr. Lynd and applying the alternate formulas I have suggested we find that heat-pipes would be as follows:

Room	Pipe Area	Pipe Diam.	Length Horizontal	Add for Length	Actual Diam.
Living Room	.851	13	4	0	13
Den	.416	9	9	1	10
Dining-Room	.361	9	8	1	10
Pantry	.163	6	14	1	7
Hall	.851	13	3	—	13
Chamber 1 ... } ... Dressing-Room } ...	.398	9	8	1	10
Chamber 2	.244	7	4	—	7
Chamber 3	.321	8	19	2	10
Bath	.093	5	2 ells	2	7

In practice the Living Room would be supplied by two 10-inch diam. pipes and the hall would probably be two 9-inch or 10-inch diam. pipes, while the upper floor would be supplied with

Room	Cellar Pipe Diam.	Wall Stack Dimensions
Chamber 1 .... Dressing-Room	12	5½ x 14
Chamber 2	8	3½ x 10
Chamber 3	12	5½ x 14
Bath	8	3½ x 10

Using large cellar pipes to connect with wall stacks to upper floor, which more closely conforms to standard good practice for 140 deg. register temperature.

In computing the size of furnace required, it is advisable to add to the heat loss from the heated portion of the building a percentage to cover loss of heat through basement heat-pipes, as a margin of safety, etc.

For recirculating systems this percentage should be from 45 per cent. to 50 per cent. of the heat losses from heated rooms.

For the building we would then require a furnace capacity of 83,566 x 1.45 = 121,170 B.t.u. or an 8,500 B.t.u. efficiency furnace of 2.376 sq. ft. grate area or 21-inch diameter grate, if combustion rate is not to exceed 6 lbs. coal per sq. ft. grate area.

This would usually be rated as a 24-inch (top diam.) firepot furnace.

For all outside air supply the furnace capacity should be 2.3 times the heating requirements of the rooms when the register

temperature is to be 140 deg., therefore for this building we would require

83,566 x 2.3 = 192,202 B.t.u. furnace capacity or 3,769 sq. ft. grate area of 26½-inch diameter grate in a furnace of 8,500 B.t.u. efficiency.

This would be a 29-inch (top diam.) firepot furnace.

Furnaces of less capacity than 29-inch diameter firepot do the work when installed with outside air supply, because (as I stated at the beginning of this discussion) the register temperature in zero weather is always considerably above 140 deg., no matter how large the piping.

With a register temperature of 180 deg. the required furnace capacity (including allowance for heat lost in transit from furnace to registers) would be about 1.9 the heat loss from the rooms which in this case would require a furnace with 24-inch diam. grate (usually catalogued as 26-inch diameter firepot). This is the size furnace commonly used for a so-called good job of all the outside air supply—leakage exhaust type of installation.

Mr. Lynd's concluding deduction that the same size heater is required for the various types of installation is not shared by the furnace industry, which has developed furnaces of liberal air passing capacity for recirculating systems, in contrast with the small air space furnaces commonly used in districts where all the air supply is taken from outdoors.

CCCLXXXI

CRUDE OIL AS FUEL IN LOW PRESSURE HEATING SYSTEMS.

BY H. S. HALEY, MEMBER

The writer desires to present to the Society, results of various crude oil fuel tests, under cast iron sectional and water tube boilers in low pressure heating systems, in the vicinity of San Francisco, California, during the last three years, and a description of the construction of the mechanical rotary burners developed during that period.

The petroleum production in the State of California as compiled by the Standard Oil Company for the period ending September 30, 1914, averages 289,979 barrels per day which, during the month of October, sold to the consumers at 77 cents a barrel.

Crude oil is unquestionably the fuel of to-day on the Pacific Coast, in regard to economy, ability to carry overloads and meet almost instantly wide ranges of loads, boiler and furnace efficiency, absence of smoke, soot or ashes, no loss or deterioration, when stored for any period of time and decreased cost of fuel handling.

The following table shows the specific gravity, weight, and heat value of California oils.

Degrees Baumè*	Specific Gravity	Weight per Bbl. in lbs.	B. T. U. per lb.	B. T. U. per bbl.
10	1.0000	350.035	18,280	6,398,600
11	0.9929	347.550	18,340	6,374,100
12	0.9859	345.100	18,400	6,349,800
13	0.9790	342.680	18,460	6,325,900
14	0.9722	340.300	18,520	6,302,400
15	0.9655	337.960	18,580	6,279,300
16	0.9589	335.650	18,640	6,256,500
17	0.9524	333.370	18,700	6,234,000
18	0.9459	331.100	18,760	6,211,400
19	0.9396	328.890	18,820	6,189,700
20	0.9333	326.690	18,880	6,167,900
21	0.9272	324.550	18,940	6,147,000
22	0.9211	322.420	19,000	6,126,000
23	0.9150	320.280	19,060	6,104,500
24	0.9091	318.220	19,120	6,084,400
25	0.9032	316.150	19,180	6,063,800

*10 deg. Baumè equals the weight of water.

The specific gravities, weights, and heating values, as tabulated above were taken from the transactions of the American Society of Mechanical Engineers, Volume 33. The calorific values were determined by means of an Atwater bomb calorimeter, and the specific gravity by means of a Westphal balance or a pycnometer flask at about 63 degrees Fahr. Water was determined by distillation.

The computed values of B.t.u. per pound, and the specific gravity, is given as well as the weight per barrel, and B.t.u. per barrel. From the table is readily seen that the average heating values of a pound of oil increases as the specific gravity diminishes and does not increase so rapidly as the weight per unit of volume diminishes. The heating value per barrel of the heavier oils is therefore greater than that of the light oils.

COMPARISON OF COST OF OIL, COAL, AND WOOD, FUEL IN SAN FRANCISCO, FOR THE PAST SEVEN YEARS

Year	Average Price Fuel Oil at Tank, per bbl.	Price Coal per Ton Delivered	Price Wood per Cord Delivered
1908	$1.10	$11. to $12.	$13. to $14.
1909	1.10	11. " 12.	13. " 14.
1910	1.03	11. " 12.	13. " 14.
1911	.65	11. " 12.	13. " 14.
1912	.65	11. " 12.	13. " 14.
1913	.75	11. " 13.	13. " 14.
1914	.80	11. " 13.	13. " 14.

COMPARISON OF EVAPORATION WITH COAL AND OIL AS FUEL.

The equivalent evaporation in horizontal tubular boilers per pound of coal, may run from 5 pounds as a minimum to 11 pounds or higher, while the evaporation per one pound of oil, may run from 12 to 16 pounds. Coal obtainable in San Francisco, gives an average, equivalent evaporation of about 8 pounds of water, per pound of coal.

Assuming the equivalent evaporation of coal and of oil to be respectively 8 and 12.6 pounds, one pound of coal will be equivalent to .625 pound of oil, or one ton (2,000 lbs.) of coal will equal 1,250 pounds of oil (or 3.71 bbls. of 42 gallons per barrel).

Crude oil is stored for low pressure heating systems, in cylindrical tanks of ¼-inch steel with seams single riveted or welded, and from 1,500 to 2,500 gallons capacity. These tanks are placed under the sidewalk with or without concrete walls, depending upon the condition of the soil, and judgment of the City Fire Marshal, who passes upon each installation. Fig. 1 shows several standard methods of the installation of oil tanks, within the City and County of San Francisco.

FIRE UNDERWRITERS' APPROVED METHOD OF INSTALLING FUEL OIL TANKS IN SAN FRANCISCO, CAL.

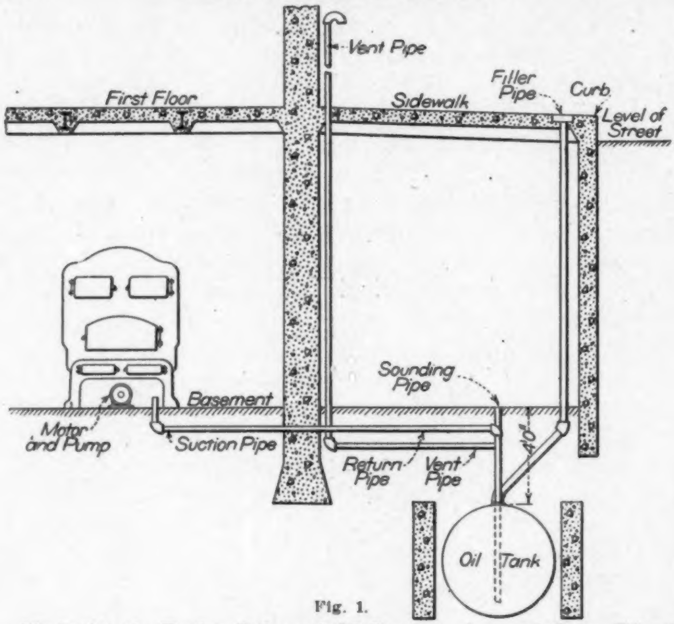


Fig. 1.

Illustrating the Heating Boiler and Oil Burner on basement floor. Sidewalk area excavated to curb line. Oil tank buried 4 feet.

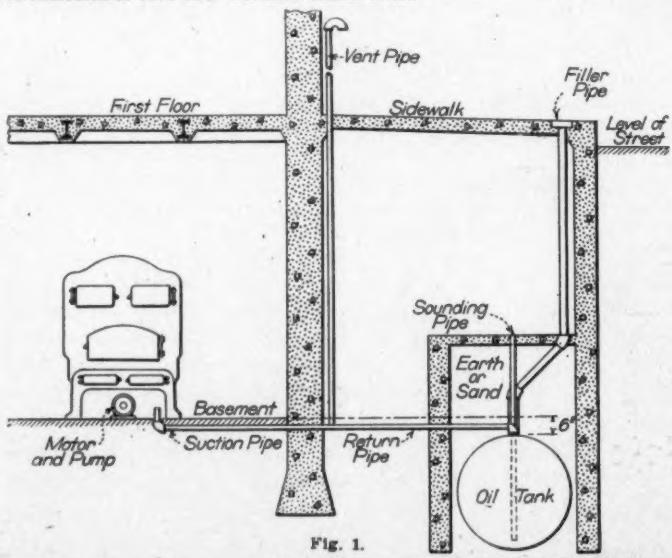


Fig. 1.

Illustrating the Heating Boiler and Oil Burner on basement floor. Sidewalk area excavated to curb line. Oil tank 6 inches below floor line. Oil tank covered with 4 feet of earth.

FIRE UNDERWRITERS' APPROVED METHOD OF INSTALLING FUEL OIL TANKS IN SAN FRANCISCO, CAL.

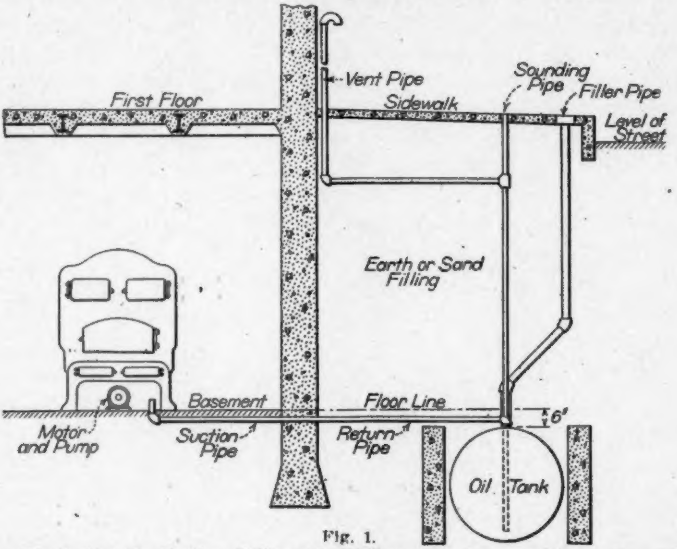


Fig. 1.

Illustrating the Heating Boiler and Oil Burner on basement floor. Sidewalk area unexcavated. Oil tank 6 inches below floor line.

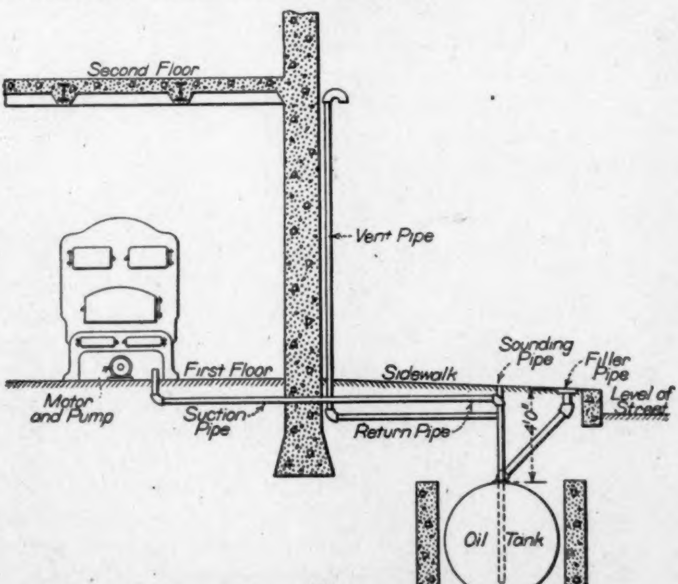


Fig. 1.

Illustrating the Heating Boiler and Oil Burner on first floor. Oil tank buried 4 feet below sidewalk at curb line.

The burning of crude oil as a fuel under cast iron sectional boilers for heating purposes was first undertaken in this vicinity by Mr. Gearhardt of Fresno, California, about 1900, using the low pressure straight shot burner, and utilizing the principle of atomization of crude oil by compressed air. The troubles arising from this system were the noise of combustion, the leakage of oil into the furnace, and the breaking of sections of the boilers. In some instances the writer has had experience in the Schools of the City and County of San Francisco, where the class rooms over the boiler room could not be occupied, because of the excessive noise due to combustion.

The first mechanical rotary crude oil burner, utilizing centrifugal force for the atomization of the crude oil, was patented by Mr. Fessler of Berkeley, California, in 1910, and was placed on the market of San Francisco in 1912.

Mechanical atomizing burners consist of a motor driven burner head to which the oil is fed, and which throws the oil off into the furnace by centrifugal force in a finely divided state. The oil pump is operated from the main motor shaft, and the whole apparatus is mounted on a single heavy cast iron bed plate. The apparatus is noiseless in operation, and is entirely automatic, being provided with no-voltage release switches, and automatic arrangement for controlling the oil supply to the burner, from the pressure in the boiler, and is so arranged that the fire cannot be extinguished by its operation. These burners are capable of operating with a clear white fire, free from smoke, within ten minutes after the fire is started, and they operate continuously without the formation of carbon in the furnace.

Complete dismantling and the entire removal of the burner and apparatus from the boiler fire box can be made in from 15 to 30 minutes from the time the fire is extinguished.

During the last two years, a burner similar to the Fessler burner has been placed on the market and is called the "Simplex Centrifugal Rotary Burner," and within the last six months a third type has appeared and is called the "Straight Shot Centrifugal Rotary Crude Oil Burner," manufactured in San Francisco.

In the Fessler and Simplex apparatus the burner head rotates on a vertical shaft driven by bevel gears from the motor shaft, while in the straight shot apparatus, the burner head is mounted directly on the motor shaft and revolves in a horizontal position.

The tests hereinafter described have, in a general way been made in the following manner. The burners are tested for different loads, and in making these tests the oil used by the burner is weighed on

a platform scale in a suitable tank. The water used is measured by a Worthington piston meter, and its weight figured from the weight of a cubic foot of water at the temperature of the feed water. The stack gases, feed water, and oil temperatures, are taken by thermometers. The steam is disposed of by blowing directly to the atmosphere thus operating the boiler at atmospheric pressure.

FESSLER BURNER

Among the first evaporative tests made were those on the Fessler burner at the Lincoln School, San Francisco, on May 11, 1912. The burner was installed under an Ideal Sectional boiler S-36-8, and was tested by W. E. Leland, a Consulting Engineer, assisted by H. L. Delaney. On May 13, 1912, in the same boiler and with the same burner, the writer, at that time Mechanical Engineer for the City Architect's office of the City and County of San Francisco, assisted by H. L. Delaney, made a second test. Both tests are tabulated as follows:—

	May 11th	May 13th
Duration of test in hours	1.55	2.
Average Temp. feed water, deg. F.	63	63
Average gauge pressure, pounds	3.5	3.
Factor of Evaporation	1.158	1.156
Oil consumed, pounds	125	135
Actual water evaporated, pounds	1350	1557
Actual water evaporated per pound oil	10.8	11.53
Equiv. evap. from and at 212 deg. F.	12.5	13.32
Developed load in square feet	3240	3600
Per cent. of rating	88.5	98.
Motor, horse power	1/3	1/3

The following is the result of an evaporative test made under an Ideal Boiler S-36-6, with the Fessler burner, in April of 1914:—

Type of fire	Low	Medium	Heavy
Duration of Test, hours	2	3	3
Average Temp. feed water, deg. F.	59	59	60
Average Temp. Stack, deg. F.	490	580	630
Average Gauge Pressure, pounds	0	0	0
Factor of Evaporation	1.159	1.159	1.158
Oil consumed, pounds	78	167	184
Actual water evaporated, pounds	945	2043	2243
Actual water evaporated, per pound oil	12.11	12.24	12.19
Equiv. evap. from and at 212 deg. F.	14.025	14.186	14.116
Developed load in square feet	2181	3159	3465
Per cent. of rating	83.4	120.	132.

REMARKS:—A trace of carbon was visible in the low and medium fire tests. No smoke or noise of combustion was noticed in any of the tests. The burner had a maximum range of 1,279 square feet, without smoke or carbon.

The results of an evaporative test made under a 110 Horse Power Badenhausen water tube boiler, with a Fessler burner, at Stockton High School, Stockton, California, in October of 1914, are tabulated below:—

Type of fire	Low	Medium	Heavy
Duration of Test, hours	2	2	1.19
Average Temp. feed water, deg. F.	68	68	68
Average Temp. Stack, deg. F.	330	334	339
Average gauge pressure, pounds	0	0	0
Factor of evaporation	1.15	1.15	1.15
Oil consumed, pounds	336	432	423
Actual water evaporated, pounds	4232	5691	5523
Actual water evaporated, per pound oil	12.6	13.17	13.08
Equiv. evap. from and at 212 deg. F.	14.49	15.14	15.04
Horse power developed	70.3	95.	140.
Per cent. of rating	64.8	86.4	127.

The accompanying table will give an idea of the actual fuel consumption of the Fessler burners, during the last year in fifteen different buildings in San Francisco.

Buildings	Number of Apts. per Bldg.	Service	Fuel oil consumed	Cost of fuel	Daily Record of Examiner Bldg. (office) (average run 20 hours daily) Service, 7800 sq. ft. steam rad. and domestic water		
					Month 1914	Bbls. per day	Cost per day
St. Elizabeth Apts.	72 rooms, 22 apts.	1200 sq. ft. steam rad. and domestic water	406 bbls. in 11 months	$345.00	Jan.	7.22	$5.77
Schnaittacher Apts.	65 rooms, 16 apts.	1000 sq. ft. water rad. and domestic water	34 bbls. in 1 month	32.30	Feb.	6.07	4.80
Otis Elevator Co. (office)		700 sq. ft. water rad.	131 bbls. in 12 months	101.35	Mar.	4.06	3.24
Whiteside Apts.	85 rooms, 38 apts.	1200 sq. ft. steam rad. and domestic water	404 bbls. in 12 months	351.48	Apr.	3.85	3.08
Hotel Baker	67 rooms	1000 sq. ft. steam rad. and domestic water	165 bbls. in 5 months	127.55	May	4.41	3.52
Moore-Watson Dry Goods Co. (warehouse & office)		4500 sq. ft. steam rad.	186 bbls. in 5 months	148.00	June	4.12	3.29
Druids Hall		1800 sq. ft. steam rad. and domestic water	60 bbls. in 4 months	52.00	July	3.46	2.76
Blackstone Apts.	72 rooms, 42 apts.	1800 sq. ft. steam rad. and domestic water	480 bbls. in 12 months	384.00	Aug.	3.02	2.41
Paris Apts.	92 rooms, 16 apts.	2200 sq. ft. steam rad. and domestic water	240 bbls in 4 months	228.00	Sept.	3.24	2.59
Solano Apts.	62 rooms, 28 apts.	1600 sq. ft. steam rad. and domestic water	416 bbls. in 14 months	354.20			
Lincoln Hotel	150 rooms, 65 baths	2200 sq. ft. steam rad. and domestic water	700 bbls. in 12 months	600.00			
Eastman Kodak Co. (office)		1500 sq. ft. steam rad.	138 bbls. in 12 months	110.00			
Inverness Apts	97 rooms, 27 apts. (2700 gal. salt water plunge)	1850 sq. ft. steam rad. and domestic water	524 bbls. in 7 months	419.20			
Holbrook Bldg. (office)	252 rooms, 2 stores	7000 sq. ft. steam rad. and domestic water	734 bbls. in 12 months	601.88			

NOTE:—Heating is required in San Francisco at all times in the year, there being no definite season.

SQUARE FEET RATING, AIR PRESSURES, AND ELECTRIC CURRENT CONSUMED BY FESSLER BURNER HEADS				
Size Burner Head	Avg. pres. inches water		Motor Consumption per hour	Sq. ft. steam rad. burner will supply
	Top fan	Lower fan		
5¾ in. Single	1.2	2.45	500 Watts	2000— 3000
6¾ in. Single	1.8	3.65	533 Watts	3000— 4500
7¾ in. Single	2.5	5.05	7 Amperes	4500— 6000
7¾ in. Double	2.25	4.5	8 Amperes	6000— 9000
8¾ in. Double	2.7	5.4	8.2 Amperes	9000—12000

Motor has a speed of 1800 R. P. M.

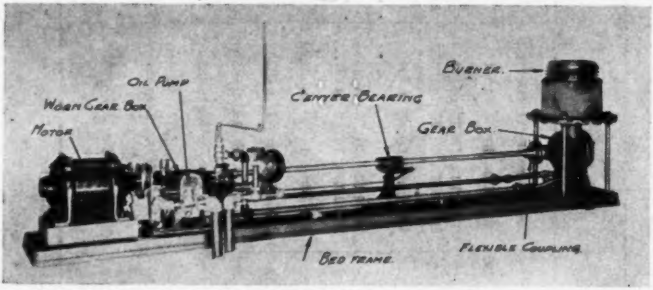


Fig. 2.

Fig. 2 shows a cut of the latest type of the Fessler apparatus and Fig. 3 shows the latest type of burner head.

The motor is mounted on wooden insulating blocks, on a heavy cast iron base frame directly in front of the boiler, and the bolts holding it in position are set in fibre insulating bushings, so that the motor is absolutely insulated from the base frame.

The flexible couplings in the shaft between the motor and the worm case, allows for perfect freedom of motion of the armature in its bearings, without binding, due to any inaccuracy in the alignment of the shaft.

The oil pump is of the twin gear rotary pattern, entirely constructed of bronze with steel driving shaft, and is driven from the main shaft by a worm and worm gear.

The gear box under the boiler is the housing for enclosing the bevel gears, which transmit the motion from the horizontal driving shaft to the vertical shaft supporting the burner head.

On the bottom of the burner head is constructed a primary air fan of the propeller type, which handles all the air supplied to the burner. At the top of the hub is constructed the two sections of

the secondary fan, which are of the multivane type, and deliver the air both above and below the oil spray.

At the bottom of the oil atomizing cup, is a trough undercut to retain constantly a thin film of oil, which by capillary attraction, connects with the film of oil fed to the burner cup from the supply

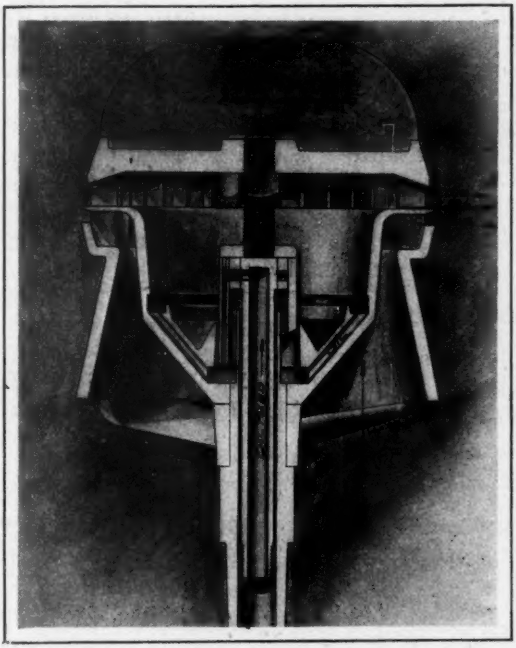


Fig. 3.

pipe in the hollow spindle. This forms a continuous ring of oil when the head is in motion, and causes absolutely even distribution of the oil over the entire periphery of the burner cup at the point where the oil leaves the head and enters the fire box. This condition is absolutely necessary for a perfectly even fire that will burn with a steady flame and without puffing. The top of the head is carried by the continuation of the oil feed spindle, and is stationary and covered with a cap of fireproof compound.

The head end of the bed plate under the boiler has four supporting legs carrying cast iron floor plates, for the support of the fire brick floor of the fire box. Immediately below the floor plates are two grooved guides, carrying an adjustable air damper, by which

the air supply to the fans is controlled, and the amount of air accurately adjusted for the fire being carried. This damper is operated by a rod extending to the front of the furnace.

The electrical starting device for the control of the motor consists of a double throw spring lever switch arranged so that in starting, the switch is thrown into the lower lugs which are connected to the line without fuses and then after the motor has reached its normal speed, the switch is quickly thrown on the upper lugs, and the motor runs on the line connected through fuses of a necessary capacity.

It is impossible to leave the switch on the lower unfused lugs, owing to its construction, which will immediately throw the switch out if the hand is removed, and the small fuses completely protect the motor from any overload that might occur when it is in operation, and solenoid will trip the switch upon any failure of the current, and the motor automatically stops.

"SIMPLEX" BURNER

Fig. 4 shows a cut of the latest type of the "Simplex" apparatus. The burner is gear driven from an electric motor installed on a

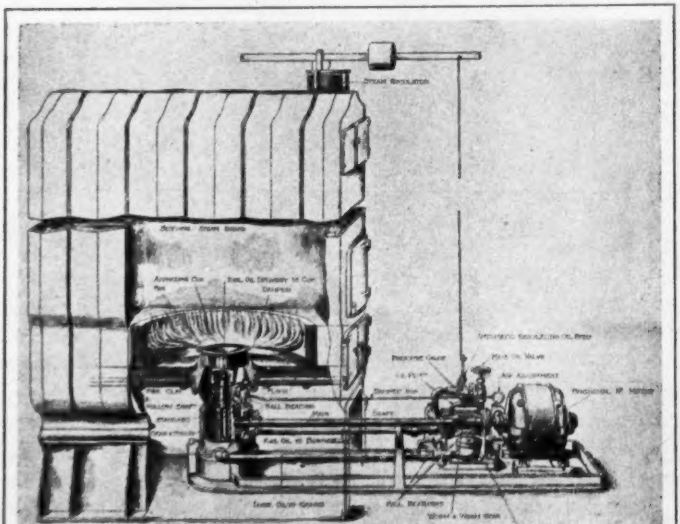


Fig. 4.

cast iron bed plate directly in front of the boiler, and all rotating shafts are mounted in the highest grade of ball bearings for end and lateral thrust.

The revolving centrifugal atomizing cup is entirely open at its discharge end, and has polished surface free of all fan blades against which heavy oil may lodge. The fuel oil is introduced at the bottom of the cup in such a manner as to be caught by the centrifugal force in a thin film, thus keeping the rotating cup insulated from the reflex heat, and as the oil climbs from the outlet in the bottom of the cup and is discharged to the fire zone, it is superheated by the reflex heat to the flash point, before it reaches the wall of the built-in fire box. The cup revolves at a speed sufficient to break up and atomize the oil, thereby evenly distributing same to the fire zone, the atomization being assisted and completed by air under pressure from a fan revolving with and under the atomizing cup, forcing the air up from below the fire box.

The fuel oil pump is driven from the main shaft by means of worm and worm gear at the necessary speeds for the different capacity of burners.

The oil line to the burner is provided with an automatic regulating lever valve, with means for attaching to a steam regulator on the boiler, for the purpose of controlling the fire so as to maintain the desired steam pressure.

The electric motor operating the pump and burner is of the 1,800 R.P.M. type, and consumes one-half horse-power for machines developing 350 to 2,868 square feet, and one horse-power for machines developing 1,440 to 9,980 square feet rating.

The motor is protected by a no-voltage release switch of an approved pattern to discontinue electric service in the event current is temporarily broken or turned off, causing the fire to go out, thus stopping fuel oil pump until fire is again lighted.

The following is the result of a test made on a "Simplex" burner during May of 1914 under an Ideal boiler S-36-6:

Type of fire	Low	Medium	Heavy
Duration of test, hours	2	3	3
Average temp. feed water, deg. F.	66	65	64
Average temp. stack, deg. F.	440	525	500
Average gauge pressure, pounds	0	0	0
Factor of evaporation	1.1525	1.1535	1.1545
Oil consumed, pounds	56	139	173
Actual water evaporated, pounds	633	1724	2090
Actual water evaporated, per pound oil	11.3	12.4	12.14
Equiv. evap. from and at 212 deg. F.	13.034	14.31	14.016
Developed load in square feet	1460	2650	3233
Per cent. of rated load	55.5	101.	123.

REMARKS:—Soft and hard carbon formed during the medium and heavy fire tests, also a trace of thin blue smoke and a slight noise of combustion. The burner had a maximum range of 1193 square feet without smoke or carbon.

Below is tabulated results of a test made under an S-36-14 Mercer boiler at the Lockwood School, Oakland, California, September of 1914, with a "Simplex" burner:

Duration of test, hours	2
Average temp. feed water, deg. F.	62
Average temp. stack, deg. F.	465
Average gauge pressure, pounds	1.31
Factor of evaporation	1.156
Oil consumed, pounds	220
Actual water evaporated, pounds	2784
Actual water evaporated, per pound oil	12.65
Equiv. evap. from and at 212 deg. F.	14.644
Developed load in square feet	6440.
Per cent. of rated load	126

REMARKS:—Very clear fire. A ½-inch horse power motor was used to drive burner. Fuel oil in storage tank very dirty and contained water.

STRAIGHT SHOT CENTRIFUGAL ROTARY BURNER

Fig. 5 is a cut of the latest type of this apparatus.

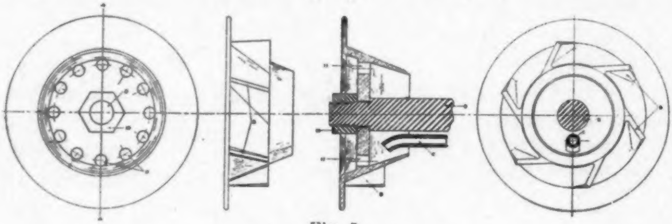


Fig. 5.
Front View. Side View. Section A-A. Rear View.

The burner head is driven direct from the shaft of the motor, and is of bronze with blades on one side for creating a forced draft. The main shaft connecting the burner head to the motor shaft is of

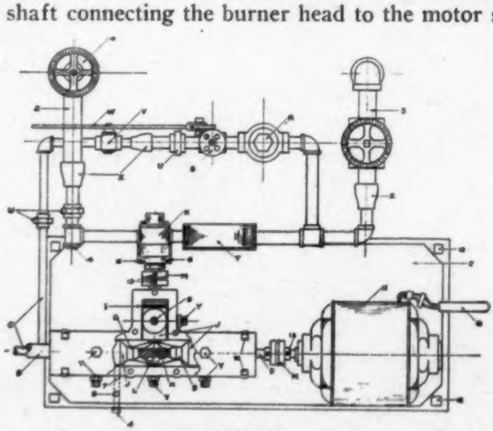


Fig. 5.
Top View.

chrome nickel steel, with a flexible coupling. About the center of the shaft from the burner head to the motor, is a worm shaft which drives a worm wheel for operating the oil pump. The burner shaft has two bearings, with ring oilers on each end, and oil grooves, which distribute the oil equally to the bearings. The bearings rest

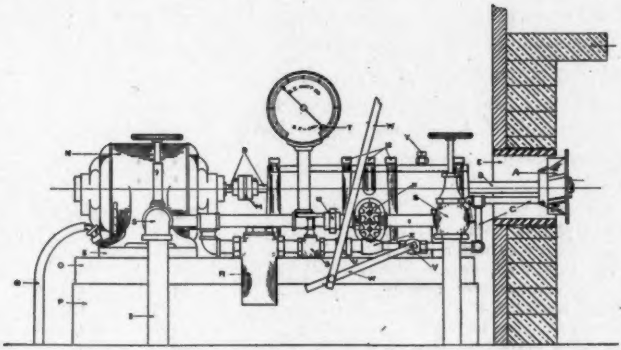


Fig. 5.
Right Hand Elevation.

on a plate, and are held in position by set screws. On the end of the shaft next to the burner is a ball bearing thrust to overcome any friction that might take place, due to end motion of the motor.

The rotary oil pump is connected to the worm wheel by a shaft, which is connected to the pump shaft by a flexible coupling. The pump shaft runs through a bronze bearing with ring oiler, and fits into the main casting similar to the main shaft bearings. All bear-

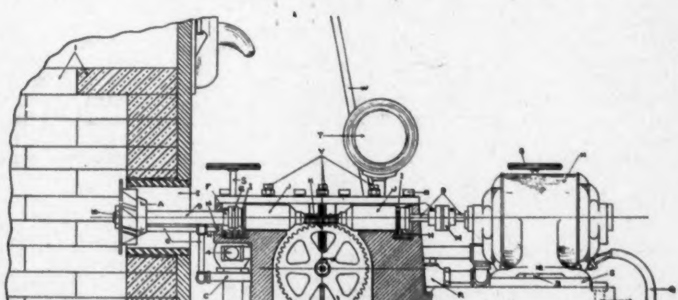


Fig. 5.
Left Hand Sectional Elevation.

ings and worm wheel are enclosed in a cast iron case containing an oil bath with a sight feed oiler on one side, showing the height of the oil in the casing.

The motor is protected similarly to the Fessler and Simplex apparatus by a no-voltage release switch, etc. The motor and burner of this type revolve at a speed of 3,500 R.P.M.

The evaporative test results, tabulated below were made under a 100 horse-power Badenhausen water tube boiler at the Stanford Court Apartments, San Francisco, in September of 1914.

This is the first evaporative test made with the burner. To date, there are only a few installations under sectional boilers, but no tests have been made.

Number of test	First	Second	Third
Duration of test, hours	1	2	1
Average temp. feed water, deg. F.	66	66	66
Average temp. stack, deg. F.	386	392	398
Average gauge pressure, pounds	1	1.27	1.5
Factor of evaporation	1.152	1.153	1.153
Oil consumed, pounds	192	457	280
Actual water evaporated, pounds	2330	5320	3100
Actual water evaporated, per pound oil	12.14	11.64	11.06
Equiv. evap. from and at 212 deg. F.	13.96	13.42	12.75
Horse power developed	78	89	103.3
Per cent. of rated load	78	89	103.3

The following evaporative test was made at the Oakland Polytechnic High School at Oakland, California, in November of 1914, under a 100 horse-power Badenhausen water tube boiler:

Number of test	First	Second
Duration of test, hours	1.15	3
Average temp. feed water, deg. F.	66	65
Average temp. stack, deg. F.	377	475
Average gauge pressure, pounds	0	48.5
Factor of evaporation	1.152	1.19
Oil consumed, pounds	312	1259
Actual water evaporated, pounds	4250	15844
Actual water evaporated, per pound oil	13.62	12.56
Equiv. evap. from and at 212 deg. F.	15.69	14.94
Horse power developed	113.5	182.5
Per cent. of rated load	113.5	182.5

TRANSACTIONS

OF THE

SEMI-ANNUAL MEETING

September 16, 1915

CCCLXXXII

SEMI-ANNUAL MEETING

First Day—Afternoon Session

Thursday, September 16, 1915

The meeting was called to order by President Kimball at 2:30 p.m., the Secretary having announced a quorum as being present.

The President: The program for the afternoon first calls for the President's address; but I am sure that in a summer meeting you do not expect anything formal. Therefore, I will just mention one or two matters connected with the interests and work of the Society. First, as I think you all know, we have been making a special effort in the matter of new members this year, and with a marked degree of success. Our Secretary, at our Annual Meeting referred to our ambition to obtain 150 new members this year; and you may recall that we have elected thus far 110. Also, we have applications from 26 prospective members now, which will give us a total, assuming these latter are elected, of 136, leaving us but 14 short of our aim at the beginning of the year.

It is suggested in connection with this matter of new members that there are a number of very able men operating large heating and ventilating plants, particularly in office buildings, hospitals and large educational institutions, and other places of that character, men admirably trained, with a wealth of practical experience and information which will be of advantage to the Society, and which would make such men very desirable members. This suggestion I think, may be well borne in mind.

I am glad to note the interest manifested in the membership of the Society. The matter of getting new members has been heretofore one of chance. This year the Membership Committee have taken hold of it splendidly, and many of you have done excellent work in bringing in new members. Some have

sent in three or four applications; many have sent in two, and many have sent in one. I therefore solicit your further efforts in this line. If possible, let us have 50 new members in the remaining three or four months of this year.

I feel that I should mention that a great deal of this increased membership is due to the active work and personal solicitation of our Secretary.

Another new step taken this year, as you are aware, is the printing of our quarterly Journal, which was undertaken with some fear and hesitation. However, it is working out splendidly, at least judging from remarks which have come to our notice all round, and the feeling of the Council itself. Also, it is paying for itself in the matter of advertisements, thus relieving us of the expense of printing our proceedings, now amounting to about $1,300 a year. There are some members of the Society who can doubtless give us considerable help in this matter of securing advertisements.

The Journal also provides a means of getting the papers before the members at an earlier date, and gives you more time to study them, so that you may be better prepared for discussion.

As the result of our increased membership, and the publication of our Journal, our finances have taken a big jump this year. Whereas we have heretofore had receipts or financial transactions amounting to approximately $6,500 a year, they are probably going to amount to $10,000 this year.

In previous years we have been one or two years behind in the publication of our Proceedings. Assuming that good fortune for which we hope, we will reach our Annual Meeting with last year's proceedings, and this year's proceedings in your hands, and, better still, with both paid for; so that for the first year in the history of the Society—certainly since the first of my connection with it, some eight years ago—the Society will be literally paid up to date. Of course, for a great many years, we have not been in debt in the proper application of the term, but we have always been behind with our bills in the sense that these Proceedings have been behind, and have been handed on to the next administration. This year we hope to get everything up to date, to pay everything, and we hope to leave a bigger cash balance than that with which we commenced the year.

There has recently been sent out to the members a questionaire relating to plant and operating cost. I suppose I have got to assume a great deal of the burden of causing this to be done, in

the face of the belief of every member of the Council that little would result therefrom. Unless something more results from it than has to date, the other members of the Council are going to be very much in the right. It is not to be expected that we will, at the first step, get all that we would like; but I do believe that with the co-operation of the members of the Society, and by spending some time on it, which I am sure our Committee and Council will be glad to do, we may really get something of worth to the Society. I have been doing some of it in my own work, especially among schools and hospitals, and it is really extremely interesting and helpful. I am sure if the members of the Society will take hold of this correctly, we shall achieve something worth while.

I have but one more word to say, and that relates to our Summer Meeting. As you know we had planned to go to San Francisco, but it appeared to be too much of a journey for our membership, and our promised attendance was very small. Therefore, we felt that "discretion was the better part of valor," and changed the place of meeting to Atlantic City. It was done on short notice, and the attendance here this afternoon is therefore extremely pleasing. Whatever we miss in point of attendance at this meeting, I hope may be made up at our Annual Meeting in January, for we are already starting the wheels in motion for the best meeting we ever had. We hope to make it best from the point of view of papers and attendance and in every possible way.

The papers which are on the program have been printed and circulated; nevertheless to bring them clearly before the meeting, they will be presented in the regular way.

In the case of papers of considerable length, I want to suggest, the suggestion being offered to me by one of those present, that the paper be presented in brief, or reviewed by the author, or his representative; believing that this plan will conduce to the best interests of the meeting.

In the program you will notice one innovation, and that is the placing of one member's name in each case after the paper, for opening the discussion. That gentleman therefore will in each case have the floor first, and after that it is open to general discussion.

I have but one more suggestion to add and that is as you discuss the paper, as one member finishes speaking, I hope he will give the right of way on the floor to as many other members as may wish to discuss the paper before he again seeks the privilege of the floor.

There is one other matter of which I wish to speak before taking up our program. There is one member of our Society who is not here this afternoon whom we shall miss. No one has done more work for the Society in the last two or three years, or contributed more for this meeting and its success. Mr. Chapman has visited Philadelphia two or three times in our interest. He has been the Chairman of the Special Committee in charge of the meeting, and has done a vast amount of work—only to become ill at the last minute—so that he is unable to attend the meeting. It may be that his work in connection with the preparations for this meeting had something to do with it; inasmuch as his cold originated on one of his trips taken in the interests of this meeting. I wish that an expression of our feelings might be sent to him as a message.

Mr. J. Irvine Lyle: I move that the Secretary be instructed to voice the sentiment of this meeting in a message to the effect that we regret very much that Mr. Chapman is not here with us, and that we hope a speedy recovery.

Mr. Chew: I suggest that this be a telegram, so that he will know that we are feeling for him.

(The motion was seconded and carried.)

The President: The first subject on our program is the Reports of the Chapter, the first being that of Illinois.

REPORT OF THE ILLINOIS CHAPTER

June 15th, 1915.

The year 1914-1915 has been a most interesting one for the Illinois Chapter. The average attendance has not been as large as some of the other years but it has averaged thirty. There seems to have been more interest manifested, as nearly every one has taken part in the discussions.

October 12 was the first meeting of the year and was the annual meeting at which the officers were elected. The result of the count of the ballots showed that C. F. Newport was elected President; E. L. Hogan, Vice-President; August Kehm, Treasurer; W. L. Bronaugh, Secretary, and H. M. Hart, William Lees and F. W. Powers, the Board of Governors. The report of the auditing committee showed the Chapter's financial condition to be excellent. A general discussion of the program for the coming year was indulged in.

During the year the Chapter participated in the display shown at a public health exhibit given by the City Club of Chicago,

covering a period from December 1, 1914, to January 4, 1915. This exhibit was to show the detrimental effect of poor ventilation and the great value of ample ventilation.

In the past the Illinois Chapter has refused to allow the press to be represented unless the representative was a member of the Society. This was rather an ancient rule and was brought about at the time of the inception of the Chapter when the Society was so afraid that the Chapter would give out some secrets over the Society's name and get them in trouble but we felt we had demonstrated our ability to preserve the integrity of the Society and it was decided to invite the press but to require them to submit copies of their reports to the Secretary for approval before being printed.

The topic discussed at the November meeting was "Smokeless furnaces, with special consideration of the down draft type." This proved to be particularly attractive to Chicago members at this time due to the requirements of the Smoke Department that all new boilers carrying a load of 1,200 square feet or over must be equipped with acceptable smokeless furnaces. This discussion, of course, brought up the question of the proper method of firing the down draft boilers.

The December meeting was devoted to a discussion of "Humidification or air conditioning for residences." It was demonstrated at this meeting that practically all of the members of the Chapter had pet schemes for curing the ills of heating systems with the application of humidifiers.

The question of the desirability of humidification came up in Dr. Hill's discussion. The principal contributors to this meeting were Messrs. Frank Douglas, A. Bement, D. I. Cooke and Professor Shepherd. The talk of these gentlemen was especially interesting as each had devoted a great deal of time to experimentation and each had a different method of procedure, but it seemed that they all accomplished about the same result.

At the January meeting the Society admitted to membership six full members and one junior. The topic of discussion was "Restaurant and kitchen ventilation." This was especially apropos as the city health department had just launched its campaign requiring all restaurant, hotel kitchens, etc., to be ventilated. Mr. J. E. Anderson, of the B. F. Sturtevant Company, was invited to be the guest of the Chapter as Mr. Anderson was one of the pioneers of kitchen ventilation in Chicago and having collaborated in the early days with the Board of Fire Underwriters and helping them to formulate their rules of require-

ment. At this meeting there were submitted drawings of the principal hotel kitchens in Chicago showing the arrangement provided for burning out of hoods and ventilating systems. It seemed to be the experience of most of the members present that the old style of standard type of fan wheel was better for ordinary ventilation than the late polyblade type, and that overhung pulley construction was preferable to overhung wheel type due to the excessive heat to which the fan wheels were subjected at the time of burning out.

The meeting of February 8th was devoted to "Trouble jobs and practical methods of correcting them." This was a subject to which most of the members present could and did contribute. It seems, however, that most of the troubles had been boiler troubles and a large number of them due to an improper consideration of the fuel.

The March meeting was turned over to a consideration of "European boiler room practice and boiler room efficiency methods of the U. S.," and stereopticon lecture delivered by Mr. W. A. Blonck. Mr. Blonck has delivered this lecture to several professional societies so it is unnecessary to give it in detail, but it was exceptionally interesting.

At its April meeting the Chapter was agreeably surprised to have the Ventilation Committee report that at last, the much looked forward to report was ready for distribution. The Society appropriated $150 for the use of the Air Washer Testing Committee, which has in the course of preparation a paper to be read before the Society and, judging by the amount of money expended on it by the Illinois Chapter, it ought to be a very valuable paper.

The evening was devoted to a consideration of "Heating swimming pools, shower baths, hot water service for hospitals, gymnasiums, etc., and the method of controlling the temperature of the water." At the Society's request we were favored with a visit from Mr. Pleins of the Clow Company who explained the method used by the Clow Company in the designs of swimming pools and shower bath equipment.

We were also favored with detailed descriptions of the methods used at the Chicago Athletic and Illinois Athletic Clubs, New York Hippodrome, Bartlett Gymnasium of the University of Chicago and several Y. M. C. A. installations, also a description of the Ford Motor Company equipment and it was especially interesting to have a report from Mr. Powers on the

new control valve which his Company had brought out for the control of temperature of shower baths and hospital service.

The May meeting was given over to a preliminary discussion of the Air Washer Committee's report. As their report was incomplete its results cannot be given here.

The Illinois Chapter is in a very prosperous condition financially.

Very truly yours,

W. L. BRONAUGH, Secretary.

On motion it was voted to receive the report and place it in the printed transactions of the Society.

ANNUAL REPORT OF NEW YORK CHAPTER

June 3, 1915.

The Secretary of the New York Chapter begs to report as follows on the year's work of the Chapter:

We have held monthly meetings from October to April, with the exception of the month of January, the meeting for this month being merged with that of the Society, which occurred in the same week.

The following topics were discussed at the monthly meetings:

October.—"Relation of Heating Buildings to Power Plants," presented by Mr. R. P. Bolton.

November.—"Test of Heating Plant of New Haven Public Library," Prof. E. H. Lockwood.

"New Rules for Designing Furnace Systems," Mr. Roy E. Lynd.

December.—"Heating the Skyscraper and its Problems," Mr. William H. Driscoll.

February.—"Air Measurement," Mr. Arthur K. Ohmes.

March.—"Wind Leakage," Mr. F. K. Davis.

The annual meeting of the Chapter was held in April, and the following officers were elected for the coming Chapter year:

President—W. H. Driscoll.

Vice-President—Arthur Ritter.

Secretary—F. K. Davis.

Treasurer—W. J. Olvany.

Board of Governors—W. F. Goodnow, W. S. Timmis, Perry West.

The annual dinner of the Cha ter took place at the Claridge Hotel, New York City, on the e ening of May 17th, at which time the new officers were introduced.

The Chapter has several committees at work, and all are taking an active interest in the work assigned to them.

Among the committees are the following:

Committee to Co-operate with the New York State Commission on Ventilation: Mr. Frank T. Chapman, Chairman; Mr. Frank G. McCann and Mr. George W. Knight.

Profession's Efficiency and Welfare Committee: Mr. Perry West, Chairman; Mr. J. I. Lyle and Mr. Frank K. Chew.

Committee to Co-operate with Mr. R. P. Miller in the framing of that portion of the New York City Building Code Relating to Ventilation: Mr. Frank T. Chapman, Mr. Arthur K. Ohmes and Mr. D. D. Kimball.

The finances of the Chapter are in a healthy condition, as evidenced by a substantial surplus in the Treasury.

The present membership of the Chapter is 76 as against 71 last year. We have added 6 new members and lost 1 by resignation—a net gain of 5 members.

Respectfully submitted,

W. F. GOODNOW, Secretary.

On motion it was voted to place the report in the printed transactions of the Society.

ANNUAL REPORT OF THE MASSACHUSETTS CHAPTER

The first meeting for the year was held Tuesday evening, October 13, 1914, for the election of officers as follows:

Andrew G. Paul—President.

Hon. Eugene R. Stone, Vice-President.

Charles Morrison, Secretary.

William T. Smallman, Treasurer.

Board of Governors—William G. Snow, Frank Irving Cooper, James W. H. Myrick.

With sixteen members we started in for business with a campaign for new members and voted to send out a special letter with the result that at our first meeting in 1915 we anticipate increasing our membership fifty per cent. by the new members accepted in our territory.

Subjects for meetings as follows:

October.—Election of officers. Beginning of campaign for new members.

November.—General Topics in Engineering. New Members.

December.—"New System of Ventilation," presented by Mr. S. H. Wheeler, Bridgeport, Connecticut.

February.—Paper presented at the annual meeting by President D. D. Kimball and Mr. George P. Palmer. A Description of the Experimental Plant of the New York State Commission with record of results of a number of Experiments made.

March.—By Mr. Laurence Franklin. Résumé of the Early Experiences of his Father, covering the installation of heating and ventilating plants.

April.—"The Present Status of the Laws of the United States on Heating and Ventilating Schoolhouses," by F. I. Cooper.

May.—Compiling list for new members.

August 24, 1914, we lost one of our Charter members, by the death of Mr. A. B. Franklin, which was a material loss to our Chapter as well as to the Society at large.

By resignation three members left us, Mr. George Huey, Mr. Charles F. Eveleth and Mr. Robert L. Folsom.

We have before us now a proposed amendment to the By-Laws, to hold the annual meeting in May instead of October.

We are sending a delegate to the Pan-American Exposition to represent our Chapter at the annual meeting of the American Society of Heating and Ventilating Engineers to be held there in September, viz., Maj. J. W. H. Myrick, and other members have signified their intention of going. Mr. Frank Irving Cooper has prepared a paper to be presented, as follows: "The Present Status of the Laws of the United States on Heating and Ventilating Schoolhouses."

Mr. L. R. Stetson has transferred his membership from Montreal and will be with us next year as well as the new members of the Society.

We close with courage, good prospects and money in the treasury.

Respectfully submitted,

CHAS. MORRISON, Secretary.

On motion it was voted to place the report in the printed transactions of the Society.

Mr. A. S. Armagnac: We have now before us the Reports of the three Chapters. The list of papers presented at the Chapter meetings is the most elaborate that I think has ever been reported from these three Chapters at any one time. If I am not mistaken the papers that have been read before the

Chapters are not preserved. However, I am in doubt about it; but so far as I know they are read and then they go back again into the hands of the men who read them.

It seems to me that the Chapters ought to make some effort to obtain possession of those papers and preserve them in the archives of the Chapter, if not in the Society. I offer this as a suggestion.

The President: The Secretary will note the suggestion. These papers, however, are often informally presented.

Mr. Frank K. Chew: The Secretary also made some remarks regarding the value of having the work of the Chapters brought into the Society in the form of papers that they might become part of the records.

The President: I do feel that I might express the opinion that the Chapter work is something of which the Society may well be proud. The Chapters are only a few years old, so to speak. They have been in existence but a short time, but all of the Chapters, and especially the New York and Chicago Chapters have splendid programs and are doing splendid work. They have good meetings, and all are backing the Society up in every way. The Annual Meetings, without the co-operation of the New York Chapter, might be a very different affair. They have done splendid work ever since their organization, and I do not think we should fail to take recognition of the effect of their co-operation.

For the last two years or so we have been working in co-operation with the National District Heating Association, through Committees, or through one Committee at least. I am sure that this co-operation has been helpful and even enjoyable. The Secretary of that Society, Mr. D. L. Gaskill, is here. They have been kind enough to arrange a meeting of their Board of Directors (I believe it is), at this time and place, to enable them to be present at our meeting and to co-operate with us as far as possible, while taking part in our meeting.

I should be very glad at this time to hear a few words from Mr. Gaskill, the Secretary of the National District Heating Association, and I am sure our members will also.

Mr. D. L. Gaskill: Mr. President, I do not know who is responsible for calling me up before your very excellent Society at this time. I happen to be in Atlantic City attending a meeting of the Executive Committee of the National District Heating Association, which we, "with malice aforethought," called at the same time as your meeting, in order that our mem-

bers who were members of your Society might take part in your deliberations as well as working in ours.

I have only to say that I am delighted to have met you, and extremely pleased to extend to you the greetings of the National District Heating Association.

The work of the two Associations is in many respects very closely allied; and one of the pleasant features of the work of our Association has been the co-operative work that has been done by a Committee from your Association in connection with one of ours. While we do not follow the same lines, and could not possibly adapt the work of our Association entirely to the work of the American Society of Heating and Ventilating Engineers, yet the two are so interwoven that I feel that our work ought to co-ordinate in many particulars, and in that way at a less expense bring the energies of both Societies to bear effectively in solving some of the problems that you have to solve.

I do not know that I can add any more, except to say that our Association is in the most prosperous condition that it has ever been in. From the reports that I hear, your Association is in a similar condition, upon which I congratulate you.

Association work, as you all know, is work that can be made very useful, or it can drag along and not accomplish what its sponsors originally intended it to do. Any Association that does not work during the twelve months of the year is not fulfilling that which its originators desired it should accomplish.

I have long maintained the idea that any Association that has only for its object to meet in an Annual Convention and hear some papers, and have generally a good time during the meeting, will not accomplish very much. An Association that works throughout the twelve months in the year, as you do through your Chapters, and as we are doing through our Standing Committees will accomplish much. There is no question about the value of an Association under that system of management.

I extend to you the greetings of the National District Heating Association, and hope your sessions will be most profitable to yourselves. As one in charge of the meetings of our Committee, I assure you that we will not allow any of our sessions to interfere with any of yours. We will meet at the odd times when you are not in sessions, and try and get some of the members of our Committee to imbibe the knowledge that I know you will put out. Gentlemen, I thank you (applause).

The President: On behalf of the Society I thank Mr. Gaskill

for speaking to us. In view of his remarks, I am going to ask for the Report of our Educational Committee co-operating with times when you are not in session, and try and get some of the Educational Committee of the National District Heating Association.

REPORT OF THE COMMITTEE IN CO-OPERATION WITH EDUCATIONAL COMMITTEE OF NATIONAL DISTRICT HEATING ASSOCIATION.

Your Educational Committee appointed to co-operate with the Educational Committee of the National District Heating Association makes a report as follows:

This Committee was not appointed until early in March of this year when a great deal of the work had been done by the Committee of the National District Heating Association, but inasmuch as four out of five of the members of their Committee were also members of this Society, this Society was well represented and its ideas worked out in co-operation.

The Educational Committee's report was read before the summer meeting of the National District Heating Association, which was held in Chicago, June 1st to 3rd, 1915, and two of the papers contained in this report have already been published in the July number of the Journal of this Society.

We would also recommend that the other part of this report consisting of "The Establishment of a Standard Coefficient for Heat Losses Effected by Wind Movement," "The Establishment of Standard Heating Elements for Cooking Apparatus with Special Reference to Low Pressure Steam," be published in the Journal at a later date, subject to the approval of the Publication Committee of the Society.

Respectfully submitted,

W. H. CHENOWETH, Chairman.
R. L. GIFFORD,
E. A. MAY,
J. J. HERLIHY,
H. C. KIMBROUGH.

President Kimball: What is your pleasure in regard to this report? I might explain that the separate papers making up the report will be presented and discussed as they come before the meeting.

It was regularly voted that the report be received and placed in the transactions of the Society.

The President: We will now hear the report of the committee representing this Society in co-operation with the National Fire Protection Association.

The report was read by the Secretary.

I beg to report as follows, on behalf of the Committee on Blower Systems for Heating and Ventilating, Stock and Refuse Conveying, of the National Fire Protection Association.

The report presented this year was a revision of last year's report containing minor changes. I do not consider them of such great importance that it would be necessary to reprint them in the 1915 proceedings of our Society.

There was, however, one important paragraph, which should be taken note of by our Society, i. e., Section 2-i, which reads now as follows: "All ducts passing through floors shall be made of or protected throughout by approved fire-resisting material, such as 4-inch brick, hollow tile, or 2-inch cement plastered partition supported by a substantial steel frame."

I take this opportunity to report that Mr. Donnelly and myself have duly represented the Society as delegates at the annual meeting of the National Fire Protection Association, and have attended practically all the sessions.

Yours very truly,

A. M. FELDMAN, Chairman.

The President: I had a notice a short time ago appointing me to serve on a committee of the National Fire Protection Association, and not being personally a member, I concluded that this Society was not backward in delegating the job to your President without previous notice. What is your pleasure with this paper?

Mr. Blackmore: I may add in further explanation of the report that at the Semi-Annual Meeting a year ago it was presented to our Society, and some of our members thought the recommendations of the underwriters were a little too onerous from an engineering point of view. Our objections to those recommendations were brought before the National Fire Protection Association and they were modified to accord with good engineering practice, that being the only change made in the Report.

Mr. Chew: I want to say a word on that report. I attended the last meeting of the National Fire Protection Association,

and I feel as Mr. Blackmore says in reference to the work of our members with the Association, that it will be necessary to watch them. Their interest is to protect the investors in Insurance Companies' Stock. I do not think they care so much about the man who owns the property or the man who occupies the building for business purposes. They do not want to lose any money by having to pay fire losses, and their restrictions about the placing of a heating plant are very apt to be detrimental to the interests of the heating trade, if their recommendations are not closely watched by a committee of this Society.

There are many exactions in the manual of the National Fire Protection Association that could be changed in the interest of the heating trade that would not increase the fire risks, and this Society should use its efforts to have these objectionable recommendations changed.

This is a good report and I move that it be accepted.

The motion was seconded and carried.

President Kimball: The next on our program is the Report of the Efficiency and Welfare Committee.

The Secretary: I received a letter from Mr. West, this morning, stating that he will be unable to get here, but he sent me his discussion of this paper.

The report was discussed by Messrs. West, Chew, Lyle, Jellett, Franklin, Kimball, Cooley, Donnelly, and Moffett.

It was then regularly moved that the President of the Society appoint a committee to continue the welfare work and if found advisable a committee to co-operate with a committee from the Chapter. Carried.

The President: The establishment of standard methods of porportioning direct radiation and standard size of steam and return mains by James A. Donnelly will now be presented.

The paper was presented by Mr. Donnelly and was discussed by Messrs. Tait, Bushnell, Kimball, Cooley, and Jellett, after which they were answered by Mr. Donnelly.

First Day—Second Session

September 16, 1915

The meeting was called to order by the President at 8:30 p. m.

The President: The establishment of a standard of transmission losses from buildings of all constructions by R. P. Bolton, will now be presented.

By request, the paper was presented by Mr. Donnelly who prefaced the presentation with some observations relative to the paper. It was also discussed by Mr. Cooley and Mr. Kimball.

The President: The next paper to be presented is that of Prof. A. C. Willard and as he is absent, the paper will be presented by Mr. Bert C. Davis.

This paper was discussed by Mr. Chew, Prof. Hoffman, (written), Mr. Armagnac, Mr. Cooley and Mr. Kimball.

The President: The next paper to be presented is a part of the report of the Chicago Ventilating Commission. It has been presented that the members may be conversant with the work that Commission is undertaking.

This paper was presented by Mr. E. T. Murphy and was discussed by Mr. Cassell, Mr. Cooley and Mr. Kimball.

TOPICS FOR DISCUSSION

The President: We will now take up some of the topics for discussion.

Do the general names of "Air Line Return System" and "Vacuum Return System" best distinguish the modern type steam systems from the common one and two pipe systems?

Mr. Cooley: On the question of Air Line Return, and Vacuum Return System, it seems a question in my mind where you distinguish between the air and the vacuum line. That is, an air line return system would be one in which the air from the radiators is removed to the return line. That is true of all such systems. When you get into the question of using the word "Vacuum," it is used rather promiscuously nowadays by most of the modern heating systems. They talk about having a vacuum system where they have no vacuum pump, and simply depend on the condensation in the radiator for a vacuum, and it is a question whether the names air return line and the vacuum return line would be used by people on distinct systems, that do or do not have a vacuum pump. It seems to me that one name of air line return system, for all systems in which the air from radiators is carried through the return before it is exhausted, would be sufficient.

Mr. Donnelly: I believe I suggested this topic for discussion, but it was changed a little from the way in which I sent it to the Secretary. I suggested the name of air return system, not air line return system. Air line return system sounds too much like air line system, and I sought to show the difference between

the older type of system, which I called the air valve system, and the newer type or air return system.

There are in general use at the present time, two types of direct steam systems; the air valve system, which requires an air valve upon each radiator, and the air return system, which does not require an air valve upon each radiator, but in which the air is carried down the return pipe together with the condensation and removed in the cellar or basement, through an open vent or air valve, or by means of a vacuum pump. The method of air removal is immaterial. The attractive feature of the new system is that no air valve is present on the radiator, to hiss, sputter and give off bad odors. It has been relegated to the cellar, where it is hoped that it will remain.

I might say that the classification of air return systems has been adopted by some of the trade papers in their directory of heating equipment, and is, I believe, to be used in the Mechanical Equipment Directory of the American Society of Mechanical Engineers.

The President: The next topic.

How Long Can Vapor-Vacuum Heating Systems be Operated as Closed Systems Without Air Binding?

Mr. Cooley: That question depends entirely, it seems to me, on the condition of system when it is closed. If it is closed at the atmospheric pressure, and then owing to condensation, and a lack of combustion in furnace, the system goes under a partial vacuum; that is one condition that can occur; of course, if that occurs, it depends on how tight the system is, how long it will be before it is air bound; as it is under a partial vacuum, the air, or course, is commencing to come in through every means it can get, and under those conditions it depends entirely on the tightness of the system.

Then, there is another condition. It might be closed when the pressure was above atmospheric, and if it closed in that condition, why, if it is air bound at all, it will be air bound at the time it is closed, the air cannot come in. No air is going to leak into the system, and if there is air in the system, that is, if it is partially air bound, if some of the radiators are not entirely turned on at the time they are closed, and remain closed, they will remain air bound until the vent is again opened, unless there is sufficient pressure to cause the air to leak out, and that is very improbable, because the pressure does not become as high as the vacuum, and the pressure is not generally over

2 pounds, whereas the vacuum may become 7 or 8 pounds. It seems to me that it depends entirely on the conditions under which the system is closed.

Mr. Donnelly: A few years ago George D. Hoffman came to the Society with a problem that if the air was expelled from a heating system, slightly above atmospheric pressure, and the outlet closed by mercury seal or check valves, that certain parts of the system quickly becomes cool when the vacuum was dropped as low as 20 inches. He at that time had vacuum air valves on each radiator. I suggested the difficulty of getting all the air out, and for a year or two afterwards conducted some experiments which have been reported in the proceedings, tending to discover how much air was expelled from the average radiator. Professor Thomas, who was associated with me on the Committee, and I had some apparatus rigged up in the office. We tried to take the air out into a bottle, and we also tried to measure, and did measure, the efficiency of the air removal by the temperature of the water of condensation. A very small portion of air left in the radiator dropped the condensation, as the air would be in the bottom of the radiator. As near as we could find out, we could not get through the ordinary air valve more than 90 or 95 per cent. of the air above the radiator; so this question involves finding out how much air can be forced out in the first place. If you only take 90 per cent. of the air out, you will have 10 per cent. in, and when you drop the pressure to 20 inches of vacuum, the steam occupies approximately three times the space, and the air occupies about three times the space. So that if you left 10 per cent. of air in, the 10 per cent. of air became 30 per eent. of air, and you would very soon have a cold spot. I think that type of vacuum air valve, is not nearly so much in use, and in fact has almost passed out of use, so that this question hardly applies so much as it did then.

I would answer by saying that this question might be preceded by one—"Can a vapor vacuum system establish a closed system with all air removed?" I might say I do not think it can, and therefore you cannot operate it for any length of time, because you cannot establish it in the first place.

Mr. Cooley: I have made some tests myself in which I obtained the same temperature at the outlet, I did in the inlet, and I could not see any difference which would indicate, according to Mr. Donnelly's theory, that we had 100 per cent. removed in that case; but I agree with Mr. Donnelly it is impossible in the beginning to ever get the complete air removal. Of course

if you did get it, and did not want to make any change in your system, you can operate, but your modulation, if you want to call it that, would be destroyed. If you shut one radiator down, you are going to effect the whole system, as the thing becomes a unit unless you have some air left in the system.

The President: The next topic.

If Operated as Open System Should the Sizes of Supply Piping and Radiation be Larger Than Required for 1 to 5 Pound Pressure Systems?

Mr. Cooley: With regard to that question I should say that they certainly do not require to be larger than what is now the practice to be used for pressure systems. It is a question whether closed systems need as large pipe as is commonly used; but I know that the results are obtained right along in these open systems where they do not use as large a pipe as is the custom to be used on the pressure systems.

The President: How about radiation?

Mr. Cooley: The amount of radiation is dependent entirely on whether or not you have a return valve at the outlet of the radiator. Take a system in which the outlet is not fitted with a valve which will close, when steam reaches it the additional radiation is placed in the radiator, simply so as to have additional condensation surface there enough to take care of any excess of steam which you might get in the radiator. You cannot regulate these supply valves for any radiator close enough to have exactly the right amount of steam coming into the radiator under actual working conditions, to heat the whole of same, and not have any go through into the return line. So the practice of most of the vapor heating companies is to add about 20 per cent. to the radiation in order to have a factor of safety to take care of excess steam. If you have a thermostatic valve on the end of the radiator, I see no reason why there should be any more radiation with a vapor system than with steam.

The President: The vapor systems do not usually use thermostatic valves.

Mr. Cooley: Some of them do. There is no reason why they cannot do so.

The President: The next topic.

In the open system should the drop in pressure in supply mains due to friction and condensation be considered in deciding the static head required to return the water to boiler against pressure?

Mr. Cooley: I hate to do all the talking, Mr. President——

The Chairman: This is a subject in which you are specially interested.

Mr. Cooley: I cannot see any reason why the drop in the pressure in supply mains should be considered in determining a static head in a vapor system which is open, as this is; why the whole system from the supply valve back to the boiler is under atmospheric pressure. It has to be. The air has a chance to get back clear into the radiator, which it does whenever you reduce the steam supply, there is always a differential on that valve from 2 to 5 or 6 ounces, according to what the system is designed for, between the pressure on the inlet side of supply valve and that in the radiator, and it is this pressure we must have in the boiler. All the static head has to do is to be sufficient to overcome the difference between the pressure in the boiler and the atmospheric pressure, so that the water in returns under atmospheric pressure, can flow back into the boiler, and the pipe friction will not have any effects on that head. It will have the effect on the pressure which you have to carry, in order to get your steam to your radiator. If you have a large pipe friction, you have to carry a higher pressure, and that will make a higher head necessary, if that is the way they want to look at it.

The President: Any other remarks on these topics? We will adjourn till to-morrow at 2:30.

(Meeting adjourned at 10:10 p. m.)

Third Session, Friday

September 17, 1915, 2.30 P. M.

The President: We will start the afternoon session with the second paper on the program, "Determination of Pipe Sizes for Hot Water Heating," by Professor F. E. Giesecke.

The President: In his absence the Secretary will present his paper.

The paper was read by the Secretary, which was followed by a written discussion by Mr. Arthur K. Ohmes.

The President: The next paper to be presented is, "Measurements of Air Flow," by Mr. Arthur K. Ohmes.

As Mr. Ohmes is unable to be present the paper will be presented by Mr. J. I. Lyle.

The paper was discussed by Dr. E. V. Hill (written) and the written reply to Dr. Hill's discussion by Mr. Ohmes and a written discussion by Mr. Ritter were also presented.

The President: Is there any further discussion to be had on this paper? I think you can all understand why we welcome, as we do, the active participation of such men as Mr. Ohmes in the working of the Society. Only illness in his family prevented him from being here to-day. I do not know of any member who has taken a better hold of the work of the Society than Mr. Ohmes.

The President: We will now present the paper, "Apparatus for the Study of Heat Radiation," by Prof. J. D. Hoffman.

In his absence the paper was presented by Mr. A. S. Armagnac.

The paper was discussed by Prof. Wm. Kent (written) and by Mr. Kimball.

The President: This morning at the suggestion of Dr. M. W. Franklin, an informal meeting was called to discuss the advisability of appointing a committee for the purpose of actively advocating greater interest in educational circles for a more extended study of the problems connected with the art of heating and ventilation and the following recommendations of the committee appointed in the morning, are herewith presented.

"WHEREAS THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has exerted an acknowledged influence for good on the practice in its special field, and those allied to it.

And whereas it is realized that there is now opening a far greater need for the continuance and extension of the services to the profession, and to mankind.

And whereas this specialized branch of engineering has not enjoyed the recognition on the part of the public at large which its importance merits.

And whereas courses of instruction in these branches of engineering are not generally available in the universities and engineering schools.

And whereas it is highly desirable for the general development of the profession, and for its greater usefulness to mankind, that entrants into these branches of engineering be as adequately prepared as are men who enter other branches of engineering.

And whereas the dignity and usefulness of any branch of professional endeavor must depend upon the thoroughness with which its practitioners are equipped to cope with the problems which arise in their respective fields.

And whereas the greatest good may accrue to the Heating and Ventilating profession, from a broad policy for the classification and dissemination of highly specialized knowledge.

And whereas this Society numbers among its membership many capable experts, whose knowledge has too infrequently been accorded the wide publicity which characterizes the common knowledge of other branches of engineering.

THEREFORE WE RESPECTFULLY RECOMMEND that the American Society of Heating and Ventilating Engineers should authorize the appointment of a Committee to present at the Annual Meeting of the Society, in January 1916, plans and recommendations for a graduate course to be given by the Society in the fundamentals of Heating and Ventilating Engineering, and the allied branches, whose object will be adequately to prepare engineers more advantageously to enter upon the practice of heating and ventilating engineering.

Submitted,

MILTON W. FRANKLIN,
FRANK K. CHEW,
JAMES A. DONNELLY,
Committee."

Considerable discussion obtained in regard to the matter during which the speakers suggested that a series of popular lectures should be given and that the committee should take up the question of compiling a text-book for the use of engineering schools and others in training the student in the principles of heating and ventilating engineering. The following gentlemen took part in the discussion:

Messrs. Kimball, Chew, Franklin, Riley, Armagnac, Nichols, McIntire, Moffett, Lyle, Cooley, Bushnell, Addams and Norman A. Hill.

At the close of the discussion it was moved by Mr. Cooley and seconded by Mr. Frank K. Chew that the President shall appoint a committee to report at the January meeting on everything involved in the suggestion.

The motion being put was unanimously carried.

The President: We will now take up the program by a presentation of the paper, "Can We Locate the Neutral Zone in Heated Buildings?" by J. J. Blackmore.

Mr. Blackmore: It might be well to say a word or two on how this paper came to be written. Some four or five years ago

I was at the top of the Singer Tower. I had been up several times before, and noticed that the draft up the tower was very strong, but this day we happened to get shut out, and it took the combined efforts of three men to get the door open before we could get down. It struck me that there was a tremendous pressure developed in heating a high building, especially in the upper portion of it, and that it would be worth while investigating the subject. I considered the matter for some time, but did little with it till Mr. Ohmes became one of the Council at the last election, and, knowing that he had done some work in that line, I put the matter up to him, with a view of getting him to start the investigation. In our talk he suggested that he would turn over his data to me, and give me any assistance I needed in preparing a paper for presentation to the Society.

I want to express my gratitude to Mr. Ohmes for the data and assistance he has given me, because without his aid I certainly could not have presented the subject as graphically as its importance deserves.

The paper was discussed by Mr. W. H. Driscoll, Mr. E. T. Murphy and Mr. S. M. Bushnell, and they were replied to by Mr. Blackmore.

A motion for adjournment till the January meeting was carried.

The President: I want to thank you gentlemen for your kindness and for your interest in this meeting.

The Convention adjourned at 5.35 p. m.

LIST OF ATTENDANTS AT SEMI-ANNUAL CONVENTION OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, ATLANTIC CITY, N. J., SEPTEMBER 16 AND 17, 1915.

MEMBERS.

Addams, Homer
Armagnac, A. S.
Blackmore, J. J.
Barr, George W.
Braemer, Wm. G. R.
Beebe, F. E. W.
Bishop, Charles R.
Boon, George
Bushnell, S. Morgan
Boyden, D.. S.
Cooley, Maxwell S.
A. E. Carpenter
Chew, Frank K.
Cassell, J. D.
Davis, Bert C.
Davis, B. H.
Donnelly, Jas. A.
Driscoll, W. H.
Franklin, M. W.
Fogg, Oscar
Goldsmith, A. P.
Greason, Samuel L.
Gilbert, Maxwell F.
Hall, A. E.
Hellerman, H. H.
Hill, Norman A.
Issertell, Henry G.
Jellett, S. A.
Kimball, D. D.
Kline, W. J.
Keeney, F. P.
Lyle, J. I.
Moffitt, W. S.
Morton, John
Murphy, E. T.
McIntire, J. F.
Mappett, A. S.
Martin, Geo. W.
Nichols, G. B.
Mellon, J. T. J.
Ritter, Arthur
Riley, C. L.
Strader, B. K.
Seward, P. H.
Timm, W. H.
Wiley, E. C.
Watson, H. R.

GUESTS.

Bristol, W. D.
Chew, Frank W.
Beatty, H. C.
Dillon, H. R.
Gaskill, D. L.
Hill, F. H.
Hetherington, Ed.
Jones, Robert Ross
Lanning, E. K.
Kingsbury, E. F.
McNair, E. E.
Monash, Louis P.
McElfatrick, J. T.
Marshall, W. J.
Plewes, S. E.
Petersen, G.
Pfeffer, H. W.
Sellman, N. T.
Shay, Frank
Tinker, Wm. E.
Wetherell, H. R.
Winterstein, C. C.

LADIES

Mrs. Homer Addams
Mrs. A. S. Armagnac
Mrs. G. W. Barr
Mrs. A. E. Carpenter
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CCCLXXXIII.

MEASUREMENT OF AIR FLOW

ARTHUR K. OHMES

In a paper read before the New York Chapter on February 15th, 1915, the author stated that the exhaustive treatment of the entire subject of air flow measurements would require a great deal more time and study than could be given at such a meeting. The Society, however, considers the paper of sufficient interest to warrant the publication in its new Journal. The author, in reading the paper, exhibited the following most accurate and finely finished instruments:

The frictionless anemometer (Figs. 9 and 10).
The A. B. C. Pitot tube (Fig. 14).
The Taylor Pitot tube (Fig. 15).
The Prandtl Pitot tube (Fig. 16).
The adjustable and portable micro-manometer (Fig. 26).

PREFACE

For convenience's sake, and so as to give the paper a well defined direction, I will divide instruments for measuring the flow of air into seven classes as follows:—

1. Direct reading velocity meters.
2. Anemometers.
3. Pitot tubes and impact discs.
4. Venturi meters and throttling nozzles.
5. Volume determination by adding heat to the air to be measured.
6. Long distance and centralized air measurements.
7. Gauges and the micro-manometer.

It might, perhaps, have been preferable to divide the classes into so-called volumetric and pressure measuring instruments,

but I believe the other classification to be superior on account of the intimate relation existing on some instruments between volumetric and pressure conditions. Each and every class of instruments described here has its proper use and application but with most of them it is difficult to secure nearly accurate results at low velocities. The accuracy of results secured by any instrument, however, must be looked at as strictly comparative, as in most other pieces of machinery. You will readily appreciate the truth of the statement that the first steam and gas engines, dynamos, radiators, boilers, or other inventions were wonders at the time of their creation and were so considered, but those inventions would be ,considered extremely crude compared with their present state of development.

It is my opinion that in the art of heating and ventilating we are only at the beginning of the development of accurate measuring instruments. Fifteen years ago we were satisfied to be able to read a column of water accurately within one-tenth of an inch. Five years ago a hundredth of an inch was considered an accomplishment, whereas to-day we can procure instruments with which we can measure a thousandth of an inch of water. Again, ten years ago we were quite proud to be able to measure an air current with some accuracy at a velocity as low as one foot per second, whereas to-day we can measure with fair accuracy a current of air of less than one-tenth of that velocity. The question now arises, do we need more accuracy and refinements? Are we to go still further in the refinements of measurements or are we returning to the old methods? There can be but one answer to this question and that is that we shall not go back, but forward, following the general movement for more exactness in all engineering and scientific work.

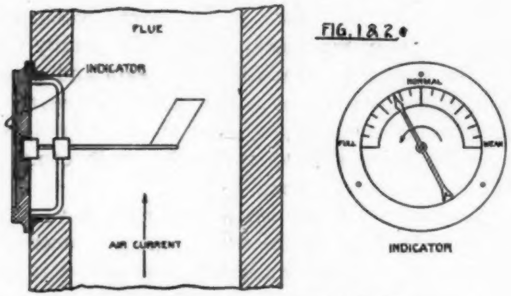
DIRECT READING VELOCITY METERS

A few sketches will give a good idea of the simplicity of these instruments. These instruments show only the velocities at the point where placed and they are not useful for accurate work. Still their use would be preferable to the usual strips of cloth or flags on registers, etc., because they will give some fairly accurate results and, incidentally, they look a little more scientific.

Figures 1 *and* 2 show a flue velocity indicator of German origin. It must be built into the flue and arranged for removal because occasional cleaning becomes necessary. Its application for class room registers may prove of value.

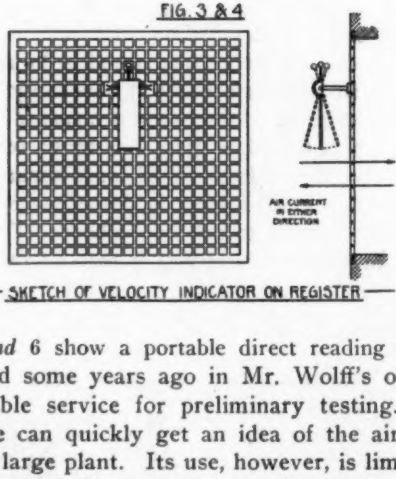
Figures 3 *and* 4 show a velocity indicator of Swiss design which may be readily applied to a register after a plant has been completed. Frequently a quadrant, calibrated for velocity readings, is attached to the upper part of the indicator; this allows the readings of velocities to be made from the register face.

FIG. 1 & 2



SKETCH OF AIR FLOW METER IN FLUE

FIG. 3 & 4

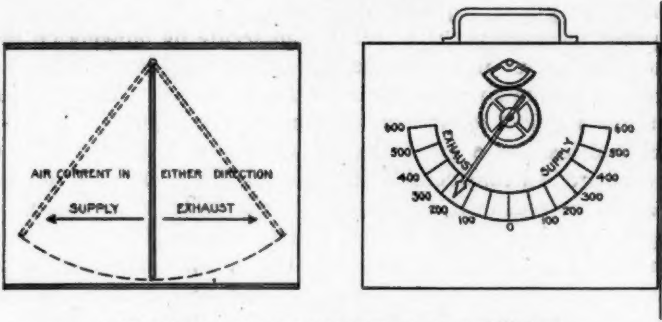


SKETCH OF VELOCITY INDICATOR ON REGISTER

Figures 5 *and* 6 show a portable direct reading velocity indicator developed some years ago in Mr. Wolff's office. It has seen considerable service for preliminary testing. With this instrument one can quickly get an idea of the air distribution even of a very large plant. Its use, however, is limited, because it must always be held vertically and for obvious reasons no exact results can be obtained with it.

Parenthetically, I wish to remark that the discussion of these instruments may be of questionable value. They represent, however, the ordinary limited capacity for understanding instruments for measuring air flow of those not technically trained but who, after all, represent the people for whom the ventilating systems are provided.

FIG. 5 & 6



SKETCH OF AIR FLOW METER FOR OUTLET

ANEMOMETERS

Many different kinds of anemometers are in use and many papers describe the accuracy to be secured with them, as well as their limitations in measuring air currents of high and low velocities. The United States weather bureau has gone quite carefully into the matter of anemometers in order to secure the most suitable instrument for measuring wind velocities. For the ordinary work required by the heating engineer the common fly wheel anemometer as shown in Figures 7 and 8 serve practically all purposes, excepting at the lower air velocities. The common fly wheel anemometer is not suited for air velocities

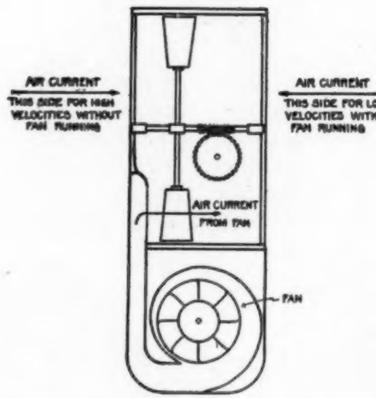
Fig. 7

Fig 8

below two feet per second for the following reason: When a wheel is turned we have frictional losses to overcome in addition to the power required for starting the wheel. The power required for starting the wheel we can secure by blowing on the wheel itself, but it is evident that work for friction must be given up by the air going through the fly wheel. This work is, consequently lost for measuring the air current itself. If these frictional losses are great, the fly wheel will only turn around when there is a considerable air velocity. Frequently the power of an air current of one and one-half feet velocity per second is required to overcome these frictional losses. If on the other hand, the frictional losses in measuring the air current are compensated for in some way the anemometer can be used for the lowest air velocities.

An anemometer which has been designed especially to overcome these objections, is shown in Figures 9 and 10. The deduction of frictional losses from the air current to be measured is

FIG. 9



secured by turning the fly wheel with an air current turned against the fly wheel itself. This air current is created by a small fan located in the box under the fly wheel. This fan is run by a clockwork spring, which will turn the fan for about six minutes and deliver air at 30 meters (about 96 feet) velocity per minute. An adjusting device allows for the calibration of the fan to secure this air velocity and the fly wheel is calibrated in absolutely quiet air. If an air current strikes the fly wheel in the opposite direction from where the fan drives it, and the fly wheel stops running, it is evident that such air current would

be at 30 meters per minute velocity. If, on the other hand, the pointer instead of showing 30 meters, shows only 20 meters, the air current counteracting the fan is at 30—20=10 meters velocity per minute.

If higher air velocities than 30 meters per minute are to be measured, the clockwork need not be used and the instrument

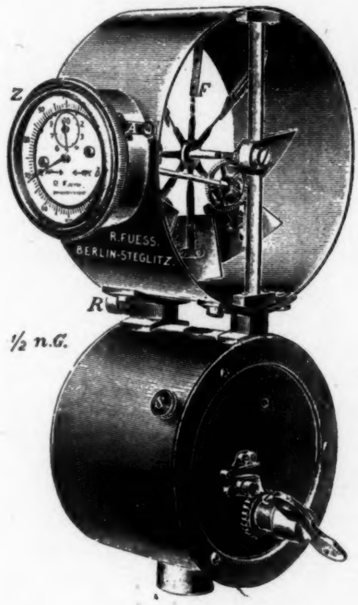


Fig. 10

may be used in precisely the same manner as the ordinary anemometer.

PITOT TUBES AND IMPACT DISCS

These instruments depend upon the condition that wherever air is in motion, it takes a certain pressure to create and to maintain such motion. In heating and ventilating engineering we deal almost invariably with air traveling in ducts and flues and we can illustrate the condition of air flow in a duct by the following figures, 11, 12 and 13. Inasmuch as we cannot see an "air column" we must employ some liquid for determining the intensity of the various pressures.

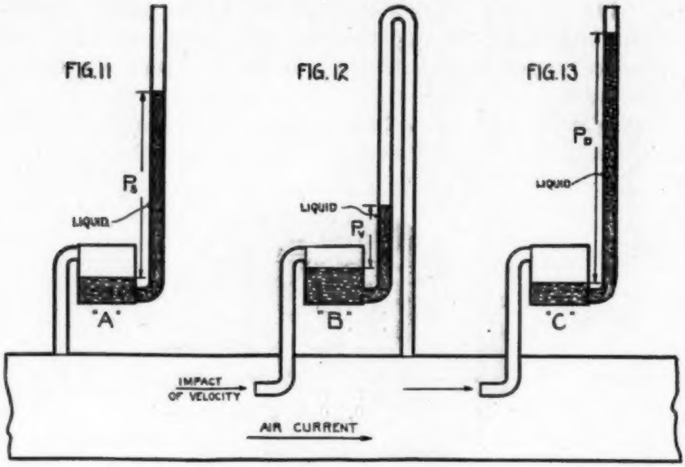


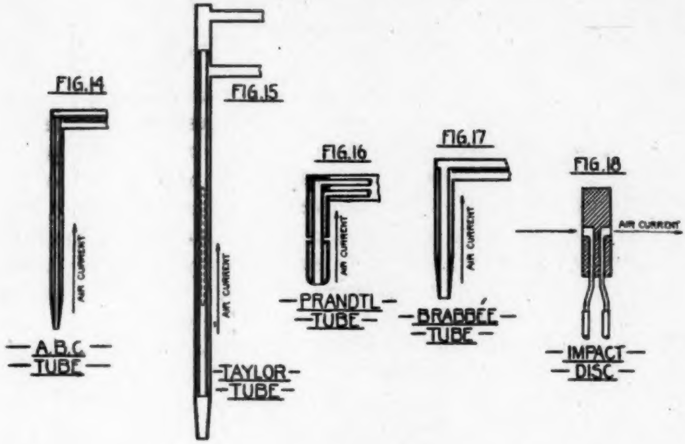
Figure 11 shows the pressure above or below the atmosphere under which the air is traveling in the duct system. This pressure may be due to many causes, of which we may mention frictional and resistance losses, bends, orifice resistance, air washers, tempering coils, and air inlets. In some cases the air may have to be introduced into a space (as in ash pits under boilers for forced blast) which must be maintained at a certain pressure. All these pressure losses and pressures added together are called the "static pressure." The Sturtevant Company has no doubt given the most suitable name for this pressure in calling it "maintained resistance."

In Figure 12 we have an illustration of the pressure created by the moving fluid and called velocity pressure. It is due to the impact of the moving air upon an obstruction placed in its path.

In Figure 13 we have an illustration of the possible highest pressure that could be measured in the duct system above the atmosphere. Here we have the sum of the static pressure plus the velocity pressure, and it is called the dynamic pressure.

To measure and record the pressures existing in a duct system in which air flows we need a specially constructed instrument, which will record all the conditions existing, and in addition a gauge or manometer is required for measuring the intensity of the pressures themselves.

Considering the first kind of instruments we have as the most



common the so-called Pitot tube, some forms of which are illustrated in Figures 14 to 17.

Figure 14 shows the American Blower Company's tube.

Figure 15 shows the Taylor tube.

Figure 16 shows the Prandtl tube.

Figure 17 shows Dr. Brabbee's tube.

Many papers can be found in various periodicals on the shortcomings and good points of Pitot tubes. A most commendable piece of work has been done by the American Blower Company in bringing their tube to a high degree of accuracy. The calibration has been evidently done with great thoroughness, judging by the description contained in a carefully prepared pamphlet published by them.

There is also a somewhat different instrument, but belonging to this class, the so-called "Impact discs" as illustrated in Figure 18. These instruments were highly thought of in Germany some years ago, but have of late been mostly displaced by the Pitot tube.

With the above mentioned instruments we can determine the velocities of the air in a measurable height of some liquid. Evidently the lower the specific weight of the liquid the greater its height. While water is by no means the most suitable liquid for this work, indeed many authorities call it absolutely unfit, it is still taken as the standard and we consequently express the static, velocity and dynamic pressure in ordinary ventilating work. in "inches of water."

Having secured the velocity pressure we can figure its velocity by considering the well known formula $v = \sqrt{\frac{2gp}{y}}$ in which p = pressure, and y = density. V, p and y must be expressed in similar terms and I have found it most desirable, in view of securing the most workable unit for the low pressures which we use in ventilating plants, to express p in oz. per square foot and y in oz. per cu. ft. To figure the velocity of an air current, if its intensity is expressed in inches of water, we should have the temperature of water, as well as exact density of the air, which latter depends upon its temperature, humidity, and the barometric pressure. In following up the density of the water or any other liquid, we will find that its temperature under ordinary conditions has but little influence. To convert the water column into pressure p consider that a layer of water of one inch height at 76 degrees exerts 5.18 pounds = 83 oz. per square foot,

consequently the formula $v = \sqrt{\frac{2gp}{y}}$

becomes $v = \sqrt{\frac{2g \times 83 \times \text{inches of water}}{y}}$

in which v = velocity in feet per second.
g = acceleration due to gravity = 32.16.
y = weight of one cubic foot of air in ounces.

If we had measured a velocity pressure of an air current with a Pitot tube of 1/20 of an inch of water we would have its velocity, $v = \sqrt{\frac{2 \times 32.16 \times 83 \times .05}{1.2}} = 14.9$ feet per second.

In which y represents dry air at 70 degrees temperature and normal barometer (29.912 inches).

As to the different tubes for determining the velocity pressure I believe the Prandtl tube to be the best one for the reason that a slight deflection should not effect the results as readily as the other tubes. On the other hand, the static pressure in the Prandtl tube is no doubt much easier affected by slight deflection from the air current than either the A. B. C. or Brabbee tubes.

It is not my intention to give a definite criticism of any tube but I wish to call your attention to the fact that if a Pitot tube is absolutely correct the velocity, as before stated, may be figured directly by the formula $v = \sqrt{\frac{2g \times 83 \times \text{inches of water}}{y}}$

If, on the other hand, a Pitot tube does not give the correct velocity it may be corrected by a constant "a" determined by experiments and the equation would be

$$v = a \sqrt{\frac{2g \times 83 \times \text{inches of water}}{y}}$$

According to a commission of the Society of German Engineers the constant for the Prandtl tube is 1; for the Brabbee tube .99 to .995. The same Commission states that the "impact discs" constant generally varies from 1.37 to 1.43 and occasionally goes as high as 1.6, which clearly shows its defects.

Now consider the Pitot tube testing results of the American Blower Company as shown on Figure 19. They were compared with the Thomas meter and it was found that the A. B. C. tube was 1.62% and the Taylor tube 14.2% below the Thomas meter readings. By careful consideration you will realize that the Thomas meter for volumetric readings should be considered correct; the fundamental principles on which it is built would make this certain. Consequently the A. B. C. tube should have a correction factor of about 1.016 and the Taylor tube 1.14.

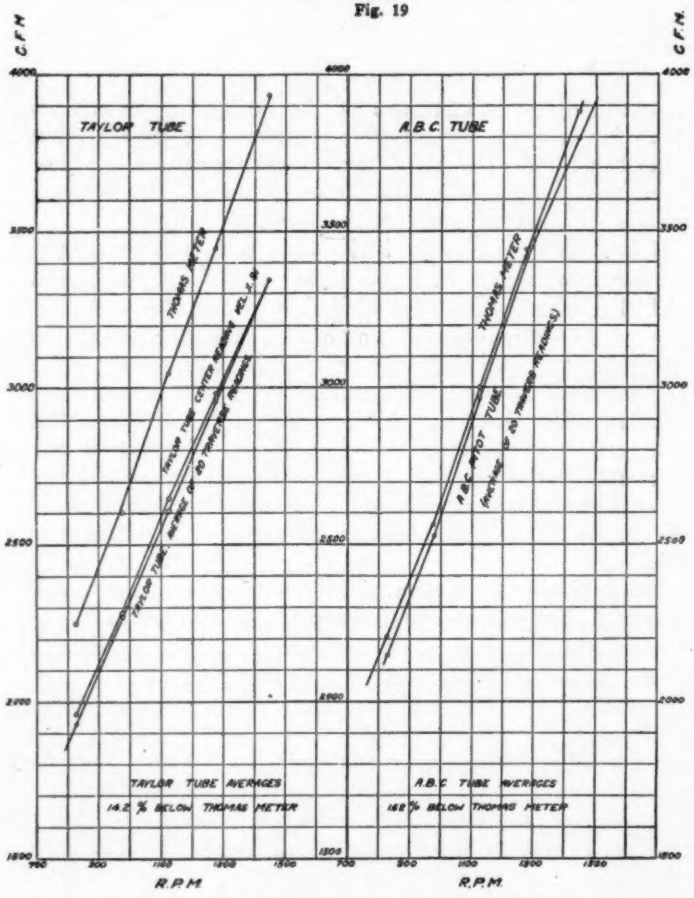
Some crude comparative tests made with the Prandtl, A. B. C. and Taylor tubes show velocity readings to vary closely as 1; 1.02; 1.14.

While this is all interesting it is of no great importance, as long as we have one accepted tube and this we have in this country in the A. B. C. tube. On the other hand, we expect to have some day an accepted standard tube the world over.

The ultimate Pitot tube, however, that should be accepted, need not be one with a correction factor of "1," in other words, one which would not require any correction factor. It may have any, or even a large, correction factor if constant for any and all velocities, provided slight deflections of the tube will not vary the results for velocity and static pressures. The error in slight deflections in most tubes seems large for one or the other condition. Similarly air currents not parallel with the walls of the ducts will make results vary greatly. Of course, in the laboratory or on a small apparatus at the factory we can create ideal conditions for testing purposes, but in general practice the matter is quite different when one considers that scarcely ever sufficient space is allotted for the ventilating apparatus. This means elbows and other conditions with eddy currents which are very unfortunate for "Pitot tube" testing. It is evident that not much consistency in air measurements with Pitot tubes in

our ordinary ventilating plants need be expected, *except when both great accuracy and integrity prevail in making the test.*

Fig. 19



Volumetric Results from Tests of "Taylor" and "ABC" Pitot Tubes against Thomas Meter.

VENTURI METERS AND THROTTLING NOZZLES

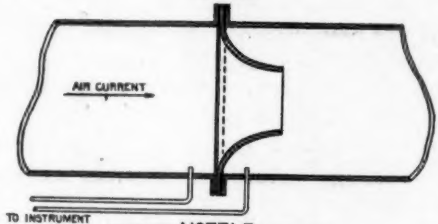
These instruments depend upon inserting into the air passage an apparatus which contracts the air passage, creating a higher velocity at a point and measuring the pressure difference due to the different velocity pressures created. In addition we have also the slight static pressure due to the resistance of the

apparatus itself. The resistance of the apparatus can be carefully calibrated and determined for certain velocities and contours and the velocity may be calculated by the formula

$$v = a \sqrt{\frac{2gp}{y}}$$

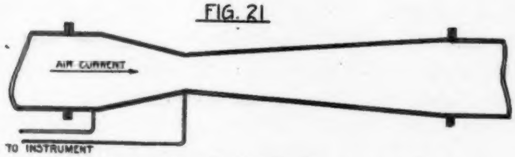
In this class of instruments belong the nozzle as shown in Figure 20 and the well known Venturi meter as shown in Figure 21.

FIG. 20



NOZZLE

FIG. 21



VENTURI METER

The value of these instruments has not been sufficiently appreciated by the ventilating engineers, yet in actual practice they are about the simplest means we have at our disposal for securing nearly accurate results. We have come to consider the venturi meter with its automatic registering devices as almost indispensable on feed water and water supply mains. Why should we not develop some thing similar for important ventilating systems? I hope to see the time when these instruments with automatic registering devices will come into more general use for keeping an easy and automatic control over the amount of air handled, and thus insure better attention than ventilating apparatus receives in general.

VOLUME DETERMINATION BY ADDING HEAT TO THE AIR TO BE MEASURED

The volume of air traveling through the ducts of a ventilating system can be approximately determined by the amount of heat given off by the tempering coil. If we know the amount of

condensation, the steam pressure, the initial and final temperature of the heated air, its humidity and the barometric pressure at the time of testing we could easily (neglecting unavoidable losses of what after all is a crude testing apparatus) calculate the pounds of air handled by dividing the total heat supplied of the tempering coil by the specific heat of the air. Evidently water, electricity or any gas whose heat capacities we can determine can also be used for the purpose of measuring the amount of air.

This principle has been utilized very cleverly by Prof. Thomas in the construction of the so-called Thomas meter. As the simplest and best heat medium he utilized electricity. For the operation and arrangement of the Thomas meter, I will refer you to the American Blower Company's pamphlet "The Pitot Tube and Fan Testing," because it describes its working and application completely.

While the instrument is most valuable for calibration and is used practically everywhere for this purpose, it has nevertheless no great practical value for the heating engineer. It is not possible to determine with it the condition under which the air moves in the duct system, that is to say the static pressure, which is usually as important to us as the volume of air itself.

LONG DISTANCE AND CENTRALIZED AIR MEASUREMENTS

In connection with this paper there is one subject about which I would like to say a few words and that is the centralizing of indicating and controlling devices for air measurements, etc., for large ventilating plants, as now used on the continent of Europe. It is an interesting development in the control of air flow measurement.

In the ordinary operation of a large ventilating apparatus, say for a large theatre, where the proper amount and temperature of air supply and the temperature of the room itself are important for comfort and health, three methods may be followed. First we will mention a case where the entire operation of radiation and dampers is left to the unskilled attendants of the theatre itself. Experience has proven that in such a case good results are seldom obtained. The second method is to provide complete automatic heat control for radiators, tempering coils, etc., and let a skilled engineer go occasionally through the building to see that the automatic control properly performs its functions. Experience has proven that with this method, using

ordinary care and common sense, very good results can be obtained. The third scheme is to have an instrument board in the engineer's office on which are placed instruments and devices to enable the engineer to see at a glance how any or all parts of the system are working, thus enabling him to detect any lowering of temperature or air supply more quickly than could the occupants of a room that might be affected. He can thus anticipate complaints by looking over his instruments (preferably sitting in an easy chair) and determining the temperature of the air supply, the temperatures that exist in the various parts of the house, the amount of air handled, etc. Surely such frequent and easy control will lead in the long run to the best results.

We can now obtain the instruments that will enable us to accomplish this result. With nozzles, venturi meters, or Pitot tubes, and manometers we may accurately determine the amount of air handled hundreds of feet away provided we have some small, inexpensive tight piping. With long distance thermometers we can determine the prevailing temperatures practically unlimited distances away. We can locate switches and operating devices for motors and dampers on the same board with the instruments and have thus not only long distance but also centralized control.

The heating engineer on the continent of Europe has pretty well succeeded in educating (not legislating) his clients to allow money for long distance and centralized control, but is obliged ordinarily to sacrifice automatic heat regulation. It is almost unthinkable for us to omit automatic heat regulation in the better work, but why should we not educate people to centralized control as well? Would we not secure better service? Could we not demand better operation? In order to give some idea of how long distance air measurements are obtained and recorded at a central station, I show in figures 22 and 23 two boards of such central stations.

The illustrations clearly show the various instruments. A list thereof, and the purpose they serve, will be found with each illustration. A little study will make everything plain and further discussion becomes unnecessary.

Fig. 22. Installed by Rietschel & Henneberg, Berlin, for the theatre in Posen, Germany. Published in the Journal of the German Congress, Dresden, 1911.

1. Voltmeter for blower.
2. Ammeter.
3. Switch for blower.
4. Rheostat for blower.

5, 6, 7, 8, 9. Long distance thermometers for: outside temperature, air supply, and inside temperatures for orchestra, three galleries, foyer and stage.

10, 11. Volt and ammeter for storage battery of current for long distance thermometers. 12. Switch for same.

13. Manometer for determining pressure in theatre.

14. Manometer for determining air volume by Pitot tube.

15. Manometer for determining total pressure difference under which blower operates.

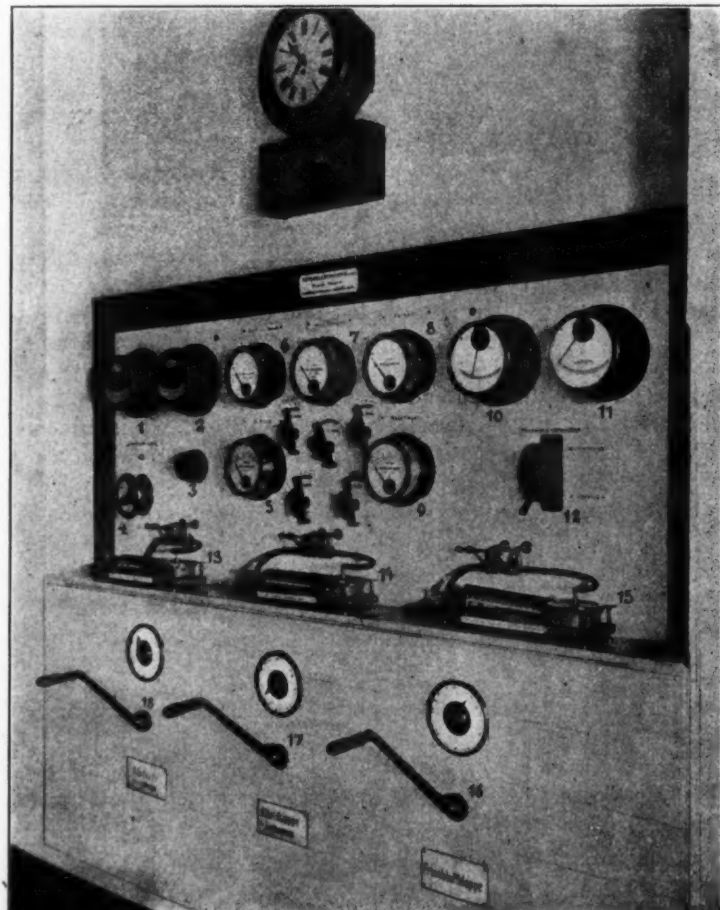


Fig. 22

16. Operating device for main air supply damper for audi-
17. Operating device for main vent damper.
18. Operating device for main air supply damper for corridors and coat rooms.

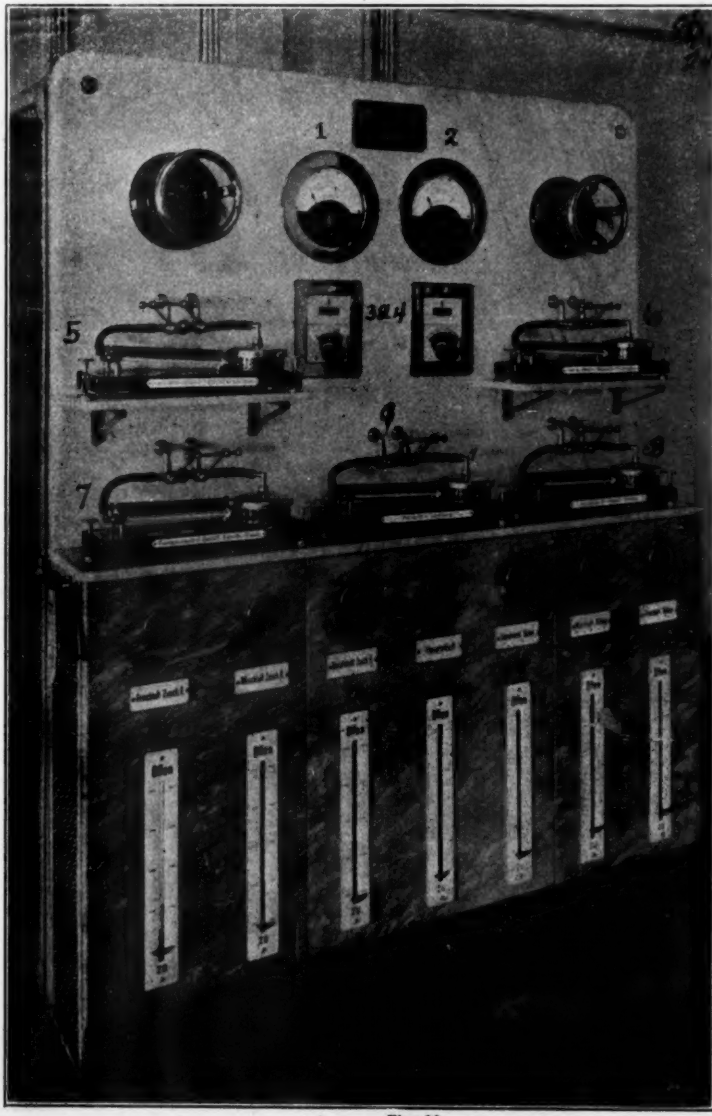


Fig. 23

Fig. 23. Installed by Rud. Otto Meyer, Hamburg, in the theatre in Cassel, Germany, by courtesy of Mr. Schiele.

1, 2. Long distance thermometers.

3, 4. Contact points for same to give different temperature in building.

5. Manometer for audience blower.
6. Manometer for stage.
7. Manometer for main duct to orchestra.
8. Manometer for main duct to stage.
9. Manometer for pressure maintained in theatre.

10 to 16. Indicators showing opening of valves and dampers for the various parts of building.

GAUGES AND MICRO-MANOMETERS

In order to secure the correct pressure exerted by the moving air, particularly the so-called velocity pressure, it is absolutely necessary to have a good gauge or manometer. An air current moving at a velocity of ten feet per second will exert only a pressure of

$$p = \frac{2g \times 83}{v^2 y} = \frac{10 \times 10 \times 1.2}{2 \times 32.16 \times 83} = .0225 \text{ inches of water}$$

On the other hand, an air current of eleven feet velocity per second (an increase of 10%) would exert under the same condition $\frac{11 \times 11 \times 1.2}{2 \times 32.16 \times 83} = .0272$ inches of water. The difference is but .0047 inches of water (less than 1/200 inch) and still if an instrument were used that could not distinguish and measure accurately this small pressure difference we will find in blower testing just 10% difference in efficiency. (Above figure at 70 deg., 29.912 inches barometer and dry air.)

On the other hand, a velocity pressure of 1/1000 of an inch of water would still create at above conditions

$$v = \sqrt{\frac{2 \times 32.16 \times 1/1000 \times 83}{1.2}} = 2.1 \text{ feet velocity per second}$$

It is evident that we not only must have accurate Pitot tubes, but also accurate manometers. For lower pressures these instruments cannot be too accurately constructed. The usual method is to employ tubes inclined 10:1. This will do for velocities above 30 feet per second, but below this point proportions of 50:1 and even 100:1 become necessary. Water cannot be used

for the latter proportions, indeed the only liquid that should be used is pure alcohol, with a specific gravity of about .83.

Two of the ordinary manometers used are illustrated in Figures 24 and 25. The adjustable micro-manometer, with quadrant

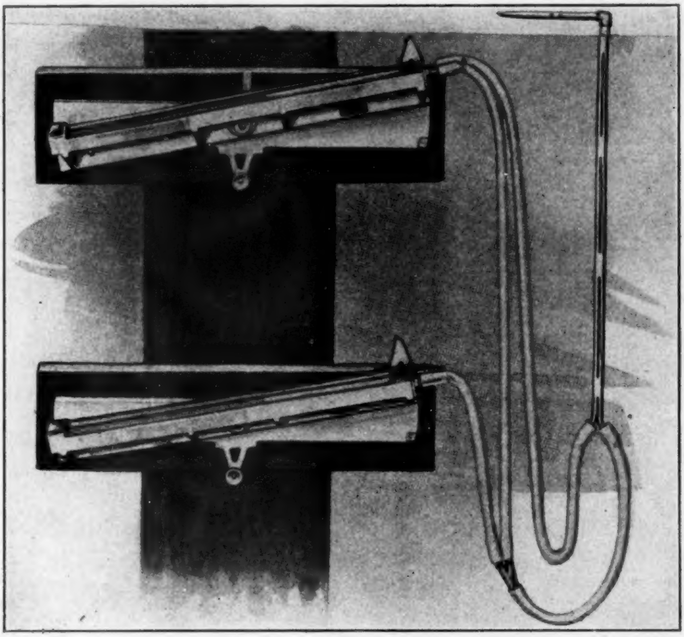


Fig. 24

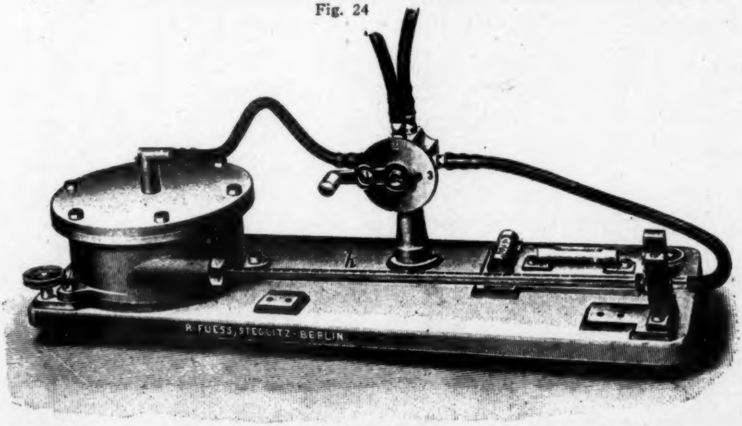


Fig. 25

and vernier for securing accurately any desired incline is illustrated in Figure 26, and this instrument is no doubt the most complete and best ever constructed.

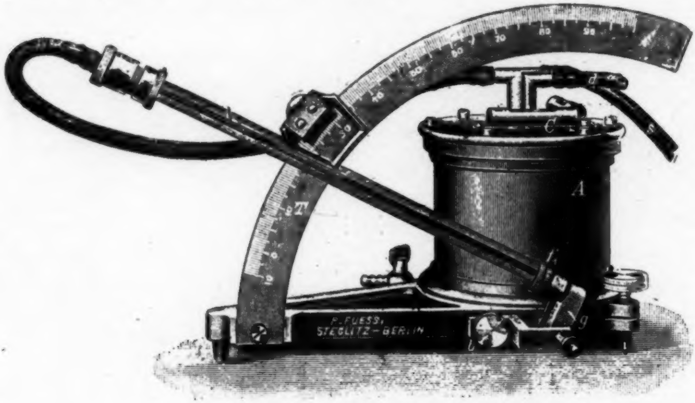


Fig. 26

The use of this micro-manometer requires that it be placed on a firm table. It is then accurately levelled up by means of two spirit levels at right angles. The quadrant is then set for the proportion wanted. The vernier on the quadrant allows for an accurate adjustment for an arc every five minutes. There is also a hand adjustment which permits the raising and lowering of the liquid in the pot.

The sine of the angle gives the proportion wanted and is to be set on the quadrant. A few of the common proportions with their degrees on the quadrant are the following:

1:1 set quadrant at 90 deg.
1:2 set quadrant at 30 deg.
1:10 set quadrant at 5° 45′.
1:50 set quadrant at 1° 10′.
1:100 set quadrant at 35′.

Any other proportion can be readily determined by the use of the trigometric functions.

ABSTRACT OF PAPER

The paper discusses the vital points of practically all known air flow measuring instruments, which the author prefers to divide into seven classes, as follows:

1. Direct reading velocity meters.
2. Anemometers.

3. Pitot tubes and impact discs.
4. Ventury meters and throttling nozzles.
5. Volume determination by adding heat to the air to be measured.
6. Long distance and centralized air measurements.
7. Gauges and the micro-manometer.

It points out forcefully that in order to secure uniform results the most accurate instruments are absolutely necessary, and that the greatest care and integrity of purpose is required to secure the proper results.

DISCUSSION

Dr. E. V. Hill: Mr. Ohmes' paper is, I think, very well written and a valuable contribution to the society. He discusses methods of making air measurements in a manner so complete and scientific that it makes me feel that I have yet much to learn regarding this important subject. I did not even know that there was such a thing as a "Frictionless Anemometer" and shall endeavor to obtain one if possible.

He describes the "A. B. C." tube as the standard in this country. While I feel that the American Blower Co. is entitled to the credit of introducing and standardizing this tube, still it will be remembered that our society has adopted a tube varying slightly from the specifications of the A. B. C. tube which we had hoped to make the standard.

Mr. Ohmes' evolution of the formula:

$$v = \sqrt{\frac{2g \times 83 \times \text{inches of water}}{y}} \text{ from } v = \sqrt{2gh}$$

is very plainly worked out but it appears to me that inasmuch as most of our engineering books give the weight of air in lbs. per cubic foot, and also inasmuch as our anemometer readings and specifications on plans call for velocities in feet per minute the formula

$$v = 1096.5 \sqrt{\frac{P}{W}}$$

is in better shape for our use than the one suggested by Mr. Ohmes. You will note from Mr. Ohmes' formula v is in feet per second while ours is in feet per minute. I have rearranged the formula slightly, squaring the constant 1096.5 and placing it beneath the radical and from this plotted a curve for determining directly the velocity from any combination of P and W that comes within the range of ordinary practice.

I am making some experiments at the present time in reading the variations of water column in a finely graduated tube by means of a Leitz micrometer eye piece used in microscopic work. This is arranged with a micrometer screw and vernier and variations in the column up to 1/1000″ can be easily determined. The difficulty in these experiments has not been in measuring the variations in the column but rather in getting a column that is quiet enough to measure. It is possible that a tube may be devised on which a certain velocity reading can be taken and the air current shut off, allowing the water column to become stationary and the difference read in this manner. This would give us a method of reading variations accurately to 1/1000″ or even less.

Regarding the use of Venturi meters and throttling nozzles, I have felt that the Chicago Chapter should take up some experimental work along these lines, as it appears that an instrument could be devised for making use of this principle.

Arthur K. Ohmes: In regard to Dr. E. V. Hill's discussion of my paper on "Measurement of Air Flow," where he advises that the velocity of the air be expressed in feet per minute instead of feet per second and the weight of the air in pounds per cubic foot instead of ounces per cubic foot, I wish to say the following.

My reason for advising that the density be expressed in ounces per cubic foot and the velocity in feet per second is mainly one for securing the best workable figures. It avoids the extreme small fractions otherwise met with.

For the practical man who must necessarily depend greatly upon his office staff the simplicity of calculations and the working with as few "fractions" as possible are always desirable. This I have found out in many years of active experience.

To prove let me say the following:

The velocity head "p" for moving an air current at 6 ft. velocity per minute equals $p = \frac{v^2y}{2g} = \frac{8 \times 8 \times 1.2}{2 \times 32.16} = 1.195$ oz. per sq. ft. Inserting "y" in pounds per cubic foot would mean that the loss would be expressed in pounds per square foot or $1.195 \div 16 = .0745$ pounds per sq. ft.

An elbow, register, louver, etc., with a constant of "1" (proportion of theoretical velocity head), would give the same values.

Now take a duct 20 ft. long by 2 ft. x 2 ft. square, in which travels an air current of 20 ft. velocity, 70 degrees temperature and normal barometer and humidity. This air current would occasion the following loss:

$$p = \frac{flp}{a} \times \frac{v^2}{2g} \times y$$

in which

p = loss in ounces per sq. ft.
f = friction coefficient = .004.
l = length in feet = 10.
p = perimeter in feet = 4.
a = area in feet = 4.
v = velocity in feet per second = 80.
y = density in oz. per cu. ft. = 1.2.
g = acceleration due to gravity = 32.16.

consequently

$$p = \frac{.004 \times 20 \times 4 \times 20 \times 20 \times 1.2}{4 \times 2 \times 32.16} = .597 \text{ oz. per sq. ft.}$$

Inserting again "y" in pounds per cubic foot would mean that the pressure would be expressed in pounds per square foot, hence there are 16 oz. to the pound it would mean

$.597 \div 16 = .0373$ pounds per sq. ft.

I prefer to work with the units of oz. per sq. ft. simply because it avoids many small fractions.

If ventilating and heating systems are figured as above it will be found to be simpler and safer in actual practice to use the oz. per cu ft.

The conversion of oz. per sq. ft. into inches of water means to divide the loss by 83; if the loss is figured in pounds per sq. ft. it means to divide it by 5.21 because $83 \div 16 = 5.2$.

Mr. Ritter: I agree with Dr. Hill in his statement that Mr. Ohmes' paper is a valuable contribution to the society, inasmuch as it deals with a subject that has had very little consideration by the average heating and ventilating engineer. Yet it is a very important part of our profession.

In this country, at the present time, only two of the various instruments mentioned are in ordinary use. I refer to the standard anemometer, figures 7 and 8. This is generally employed for measuring air velocities from 100 to 1,000 feet per minute. For higher velocities and pressure readings the pitot tube with inclined manometer are generally used, figure 24. The combination of pitot tube and micro-manometer, figure 26, would make a most accurate and complete testing outfit, but up to the present time micro-manometers are not manufactured in this country.

Mr. Ohmes illustrates the three different pressures which we find in ventilating systems. The dynamic or total pressure is the sum of the static pressure (sometimes referred to as maintained resistance) and the velocity pressure. Most of the pitot tubes used for ventilating work give approximately the same dynamic or total pressure. It is, therefore, only essential to measure one of the other two pressures accurately to procure the third. This is what the American Blower Company endeavor to do in producing an outfit which would measure accurately the static pressure.

Mr. Ohmes has shown the inaccuracy of the Taylor tube for measuring static pressures. This is due to the fact that the air velocity gets into the long static slot shown in figure 15. If you will cover the static slot in the Taylor tube with a very fine mesh copper wire gauze, you will find that the reading of the static pressure will then be approximately that of the A. B. C. tube.

Mr. Ohmes refers to the use of water as the standard of measurement, and most of the tables which are printed specify the pressures in inches of water. It has been found that the use of gasoline in the U gauges or manometers is preferable to water. When this is used, however, a correction has to be made in the reading according to the specific gravity of the liquid used. On account of the capillary action of water in the small tubes you do not get a clear line the same as you do with gasoline. Of course, alcohol is also good, but it evaporates so rapidly that it is not so serviceable as gasoline.

In the discussion of the article Dr. Hill refers to the fact that the pitot tube adapted by this Society is somewhat different than the A. B. C. standard outfit. As far as I have been able to determine, the outfits are exactly the same, with the exception that the specifications for the A. B. C. outfit call for not less than four holes in the static tube, not exceeding .02 in. in diameter, whereas the requirements of the Committee on Standardization, page 211 of our transactions, volume XX, state that there should be eight or more holes, .02 in. in diameter, an equal number on each side of a ¼ in. tube 1/32 in. thick.

The A. B. C. standard pitot tubes have been made with as many as sixteen holes for measuring the static pressure, but find that four holes give exactly the same results, which makes a greater number superfluous.

CCCLXXXIV.

THE DETERMINATION OF PIPE SIZES FOR HOT WATER HEATING SYSTEMS

BY F. E. GIESECKE,

Professor of Architecture, University of Texas

The writer believes that a considerable number of American Heating Engineers are not familiar with the accurate method of determining the sizes of pipes for hot-water heating systems, developed by Prof. Rietschel, and presented in his Leitfaden, and for that reason wishes to submit the following general description for their consideration.

The theory on which the method is based is very simple; it is this: When a hot-water heating system is being operated steadily, that is, when heat is being applied to the water in the boiler at a uniform rate and given off by the radiators at a uniform rate, the water flows through the system with a uniform velocity, and the frictional resistances in the several circuits are exactly equal to the forces which produce the flow through those circuits.

The condition of uniform flow, described above, which is made the basis of the design in this method, is to be distinguished from that which exists when the operation of the system is begun—when the water flows with an accelerated velocity, and from that which exists when the operation of the system is discontinued—when the water flows with a retarded velocity until it finally comes to rest.

To apply the method in practice it is necessary to determine the force which tends to make the water flow from the boiler to any particular radiator, called the pressure-head of that radiator, and to select the sizes of the pipe connecting the radiator and the boiler so that the friction in the pipe, fittings, etc., will be equal to the pressure-head, when the water flows with the velocity which it must have in order to deliver the required heat to the radiator.

The pressure-head, is, in every case, simply the difference in the weights of the two columns of water connecting the radiator and

the boiler; it can be very easily determined from the weights of water at different temperatures. The pressure-heads are expressed in feet of water of the average temperature of the two columns. The height used to determine the pressure-head for a radiator should be measured from the average elevation of the two connections of the radiator to the average elevation of the two connections to the boiler, or to the flow main, to which the two risers are connected.

Table I shows the pressure-heads for various heights and for various differences of the temperatures of the water in the flow and return risers.

TABLE I.

Value of pressure-head, in $\frac{\text{feet}}{10{,}000}$ for various heights and various differences in temperature of water in flow and return pipes.

Height in Feet.	Difference in temperature, in degrees Fahrenheit of water in flow and return pipes.				
	20	25	30	35	40
2	142	178	211	247	283
4	283	355	421	494	566
6	425	533	632	741	849
8	566	711	843	987	1132
10	708	889	1053	1234	1415
12	849	1067	1264	1481	1698
14	991	1249	1474	1728	1981
16	1132	1422	1685	1975	2264
18	1274	1599	1896	2221	2547
20	1415	1777	2106	2468	2830
22	1557	1955	2317	2715	3113
24	1698	2133	2528	2962	2296
26	1840	2310	2738	3209	3679
28	1981	2498	2949	3456	3962
30	2123	2666	3159	3702	4245
32	2264	2844	3370	3949	4528
34	2406	3021	3581	4196	4812
36	2547	3199	3791	4443	5095
38	2689	3377	4002	4690	5378
40	2830	3554	4213	4937	5661
42	2972	3732	4423	5183	5944
44	3113	3910	4634	5430	6227
46	3255	4088	4845	5677	6510
48	3397	4265	5055	5924	6793
50	3538	4443	5266	6171	7076

Note:—The values in this table were determined as follows:
Water at 180 degrees weighs 60.55 lbs. per cubic foot.
Water at 170 degrees weighs 60.73 lbs. per cubic foot.
Water at 160 degrees weighs 60.98 lbs. per cubic foot.

If the flow and return risers are 20 feet high and the temperatures of the water in the same 180 and 160 degrees respectively, the difference in the weights of the two columns will be 8.6 lbs. per square foot, or 8.6 per 60.73 feet of water at 170 degrees; this is equivalent to $\frac{1415}{10000}$ = .1415 feet or about 1.7 inches of water

The elevations used to determine the pressure-heads are indicated in Figs. 1 and 2; the former represents an elementary heating system and the latter one of a number of risers installed in the Academic Building of the Agricultural and Mechanical College of Texas, the heating system of which was designed according to the method to be described.

Fig. 2 also shows the pressure-heads, determined from Table I, and the friction-heads, as recorded in the original calculations for

the design. In designing the risers for this building it was found impracticable to keep the friction-head for the lower radiator as small as its pressure-head without giving that radiator a separate return riser instead of connecting it to the general return riser which serves the three upper radiators.

The return risers were connected into the top of the main flow pipe, as shown, in order that the water in that pipe might attain its average reduced temperature as quickly as possible.

The determination of the friction-heads in the several pipe-circuits is more difficult than the determination of the pressure heads; evi-

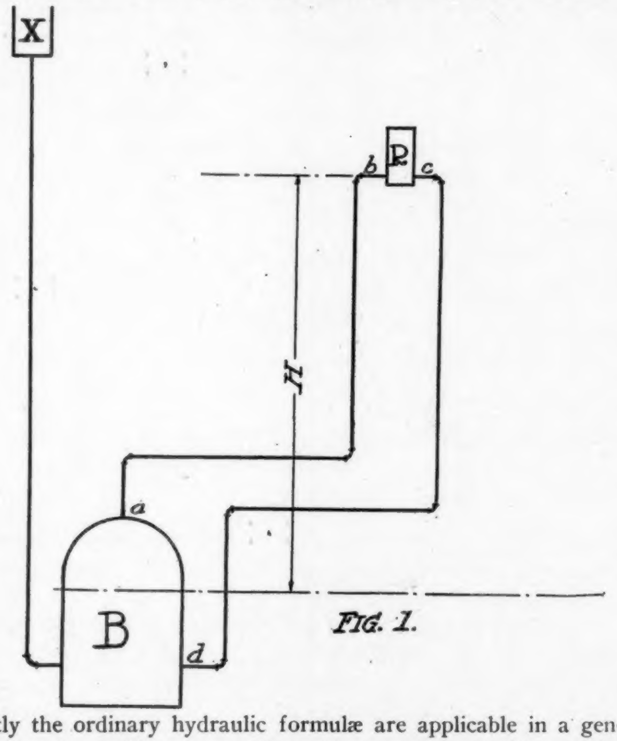


FIG. 1.

dently the ordinary hydraulic formulæ are applicable in a general way, but the problem is complicated because the velocities are lower than those commonly met with, because an unusually large proportion of the friction is due to pipe-fittings, and because the water is at a higher temperature than that with which the experiments were made which form the basis of the regular hydraulic formulas.

When Prof. Rietschel began his work he used Weisbach's formula for the friction of water in iron pipes and also his values for

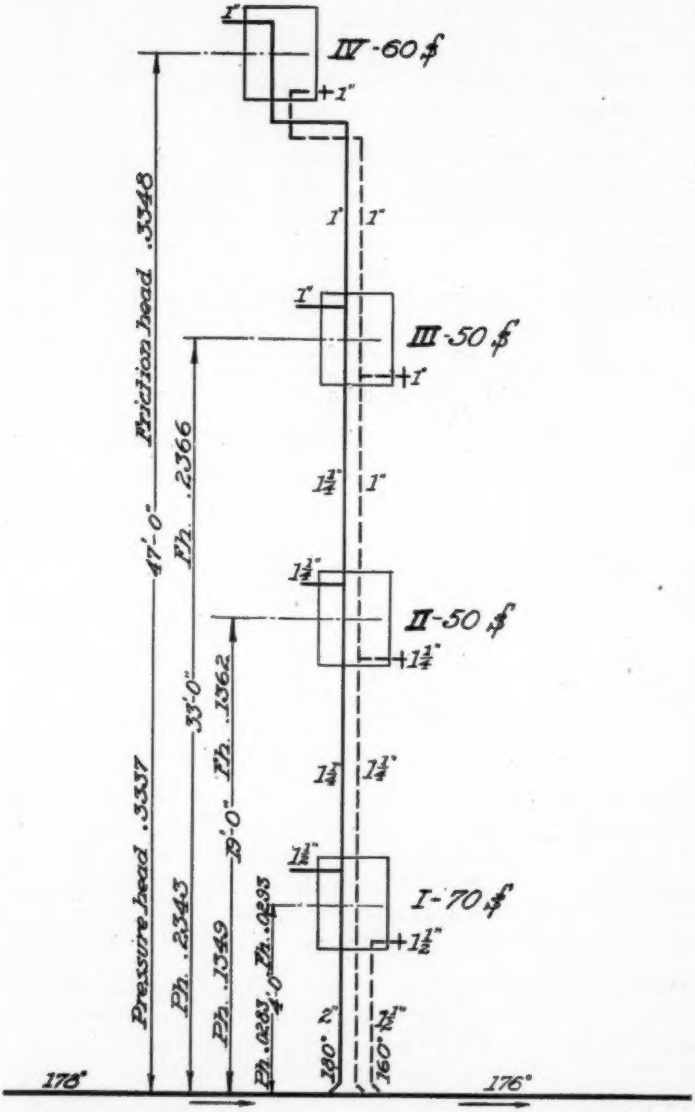

IV-60
III-50
II-50
I-70
Friction head .3348
47'-0"
Pressure head .3337
Fh. .2366
33'-0"
Ph. .2343
Fh. .1362
19'-0"
Ph. .1349
Fh. .0293
4'-0"
Ph. .0283
178°
176°
180°
160°

FIG. 2.

the friction in pipe fittings, and prepared an elaborate set of tables of co-efficients to be used in the calculations of friction-heads.

Fig. 6 shows values of the friction factors determined by Weisbach's formula $\left(.014 + \frac{.0172}{\sqrt{v}}\right)$ it also shows some of the curves constructed by the writer from data published in modern text-books on hydraulics, and successfully used by him in the design of a

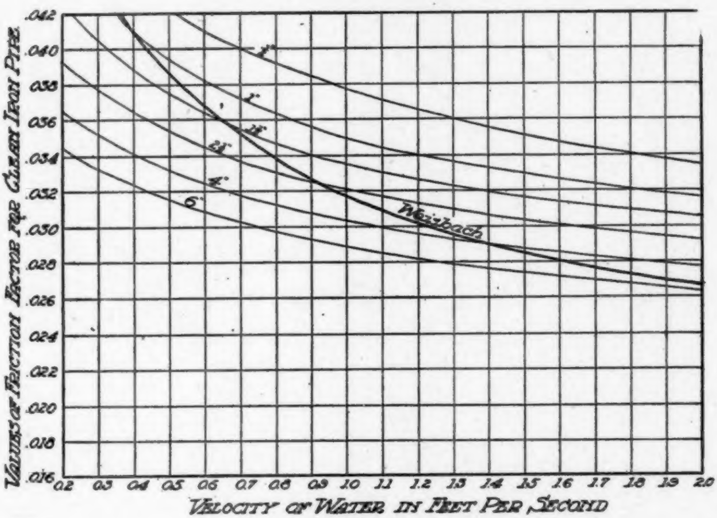


Fig. 6.

number of heating systems. These curves were used in preference to Prof. Rietschel's tables because the latter were prepared for German pipe sizes, and hence not adapted to American pipe, and because the writer felt that the friction in pipes could be determined more accurately by modern hydraulic formulas than by Weisbach's formula, in which the friction factor is independent of the diameter of the pipe.

About six years ago Prof. Rietschel began an elaborate series of experiments to determine the friction of water in pipes and pipe fittings with especial reference to its application in designing hot water heating systems.

In these experiments water varying from 50 deg. to 203 deg. in temperature and from .03 to 9.8 feet per second in velocity, and pipes varying in size from ½ to 5 inches, and in length from 16 to 65 feet were used.

The experiments were completed by Dr. Brabee, and a revised table of co-efficients based on the same published in the fifth edition of Rietschel's Leitfaden zum Berechnen und Entwerfen von Lüftungs und Heizungs Anlagen.

Fig. 7 shows a series of curves which represent, in a general way, the friction factors as determined in these experiments, and for comparison, the friction factors as found by Weisbach's formula, and shown in Fig. 6.

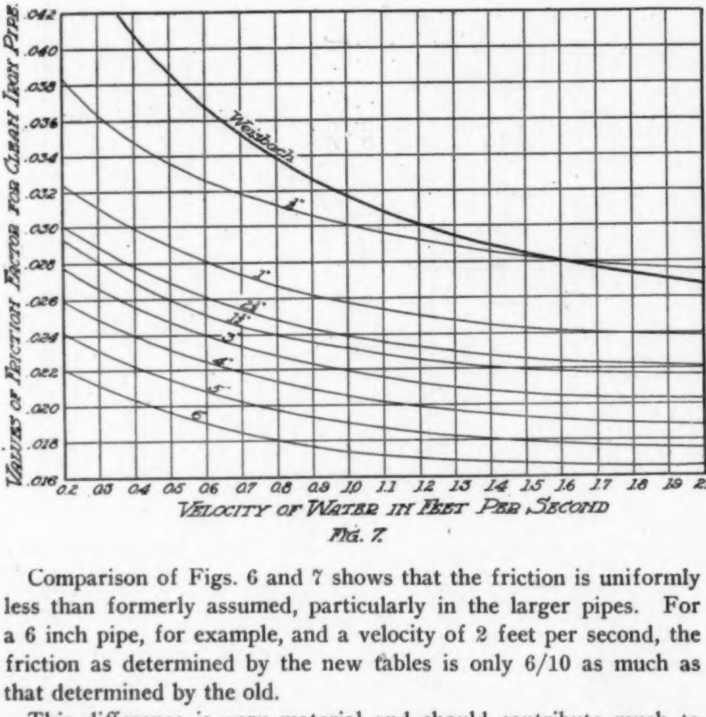


Fig. 7

Comparison of Figs. 6 and 7 shows that the friction is uniformly less than formerly assumed, particularly in the larger pipes. For a 6 inch pipe, for example, and a velocity of 2 feet per second, the friction as determined by the new tables is only 6/10 as much as that determined by the old.

This difference is very material and should contribute much to reduce the cost of hot-water heating systems, particularly in large installations and in central heating systems.

The largest system designed by the writer is the one for the Agricultural and Mechanical College referred to above, and shown, in a general way, in Figs. 3, 4 and 5.

The general arrangement of the mains, in the basement of the building, is shown in Fig. 4; it was intended that this building should be supplied with heat from a tunnel to be located in rear

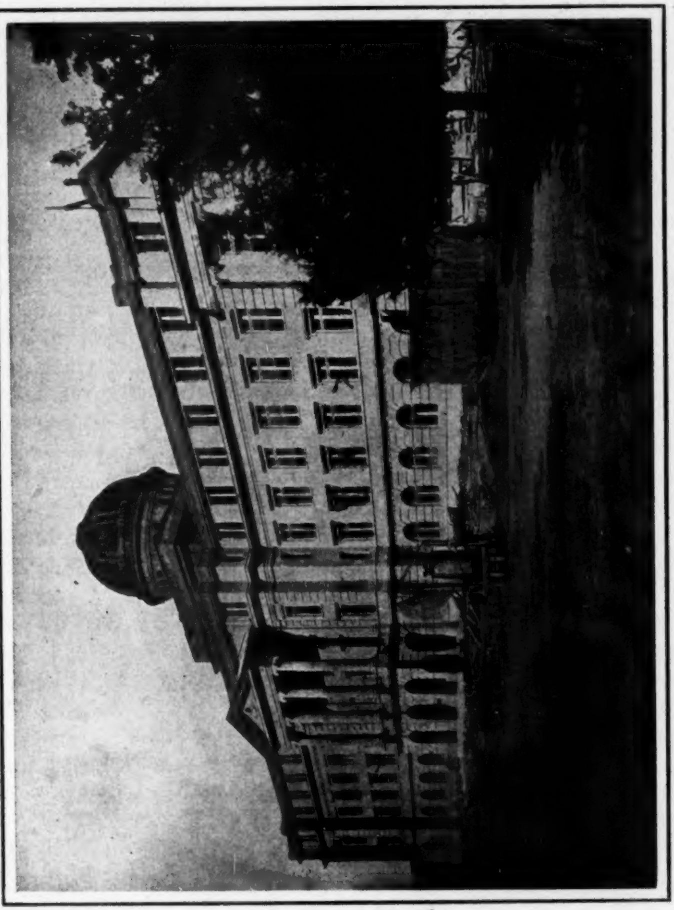


Fig. 3

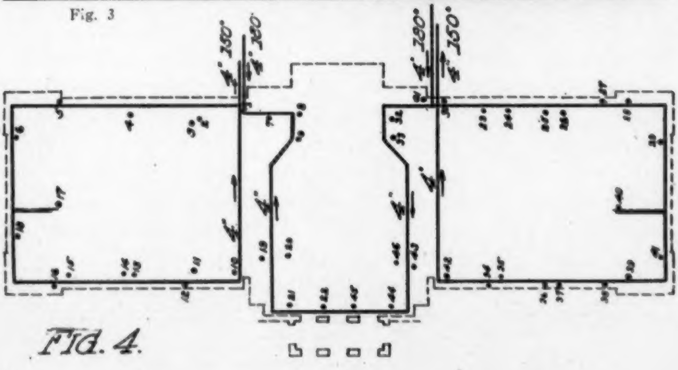


FIG. 4.

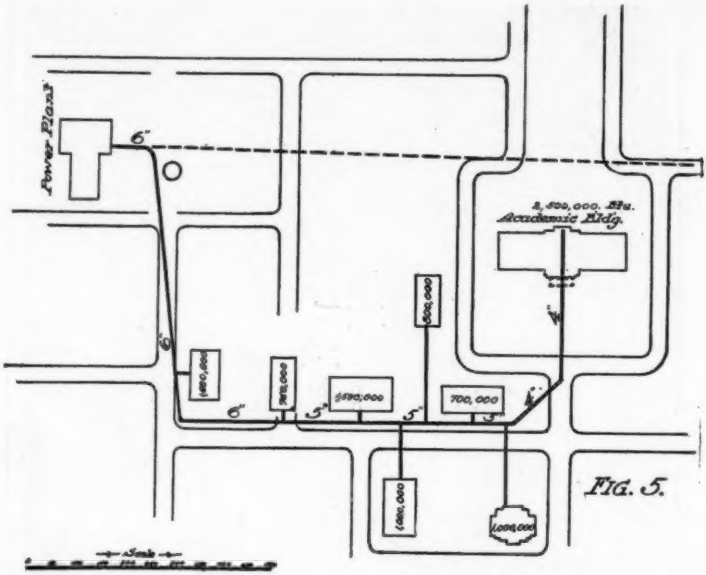


FIG. 5.

of the building as shown by the dotted lines in Fig. 5, but as this tunnel was not constructed, it became necessary to connect the building to the pipe shown by full lines in Fig. 5, which was intended only for the buildings located along the same. The water for the system is heated in the power-plant by means of exhaust steam, supplemented with live steam when necessary; the pipe is 6, 5 and 4 inches in diameter as shown, and supplies, at present, eight buildings requiring about 9,750,000 B.t.u.

When the Academic Building, with its 2,500,000 B.t.u. had to be added to this line, the friction in the pipe was calculated by the old co-efficients, and the conclusion reached was that the one circulating pump installed in the power-plant would not be able to overcome the additional load, and a second pump was installed, to be operated in tandem with the first, but the experience of the past winter has shown that one pump is able to overcome the total friction in the line; it was never necessary to operate the two pumps.

It should not be concluded, however, that the old Rietschel co-efficients are unreliable and should be abandoned, but only that, in a system designed according to the old tables, a considerable number of pipes will be larger than necessary.

That a heating system designed according to the old co-efficients will operate satisfactorily has again been testified to by Dr. Marx

in a description of the Heating System of the City Hall of one of the suburbs of Berlin, published in Gesundheits-Ingenieur, December, 1914.

The heating system in this building includes 949 radiators having, together, 62,000 square feet of heating surface, 43,500 feet of pipe ranging in size from ⅜" to 11½", the farthest riser being located 425 feet from the boiler room. The maximum temperature of the water in the flow line 194 degrees, and the minimum temperature in the return 149 degrees.

Dr. Marx reports that every radiator operated properly from the very beginning, with practically no individual adjustment, and cites this to show that installations of the largest magnitude can be designed satisfactorily according to the original tables.

To explain the practical application of the method of determining pipe sizes, let it be required to convey 250,000 B.t.u. through a pipe 150 feet long, when the difference in temperature of the water, in the flow and return pipes, is 20 degrees, and when the center of the radiator is 20 feet above the center of the heater.

Such a problem would arise when a number of radiators are located on one floor and connected to the heater by means of a single pipe.

The pressure-head producing flow is 1415 feet, as shown by Table 1, and the problem reduces itself to that of finding the size of a pipe so that the friction-head will be .1415 feet in 150, when the water flows through the pipe with the velocity necessary to convey 250,000 B.t.u. per hour.

It will generally be necessary to make several trial calculations before the correct size is determined.

Let us begin by assuming a 3-inch pipe and an average temperature of the water of 170 degrees; at that temperature water weighs 60.77 pounds per cubic foot; as the drop in temperature of the water is to be 20 degrees, each pound of water flowing through

the pipe is therefore $\frac{250{,}000}{20 \times 60.77 \times 3600}$ or, .0057 cu. ft. per second

Since 19.5 feet of 3-inch pipe contain 1 cu. foot, the velocity of the water in the pipe must be .0057 x 19.5 or 1.11 feet per second.

Having the velocity of the water, the friction-head (h_f) may be found by means of the formula, $h_f = \frac{f \times l \times v^2}{d \times 2g}$;

the value of the friction factor, f, to be used in this formula, may be taken from one of the standard text-books on hydraulics, or from the diagram of Fig. 6; for a 3-inch pipe and a velocity of 1.1

feet per second, it will be found to be about .031; by substituting in the equation and solving, we find the friction-head to be .3504 feet; since this is much more than the available pressure-head, the pipe is too small, and a similar calculation must be made for a 3½-inch and possibly for a 4-inch pipe.

To make such calculations for all pipe-circuits would be very

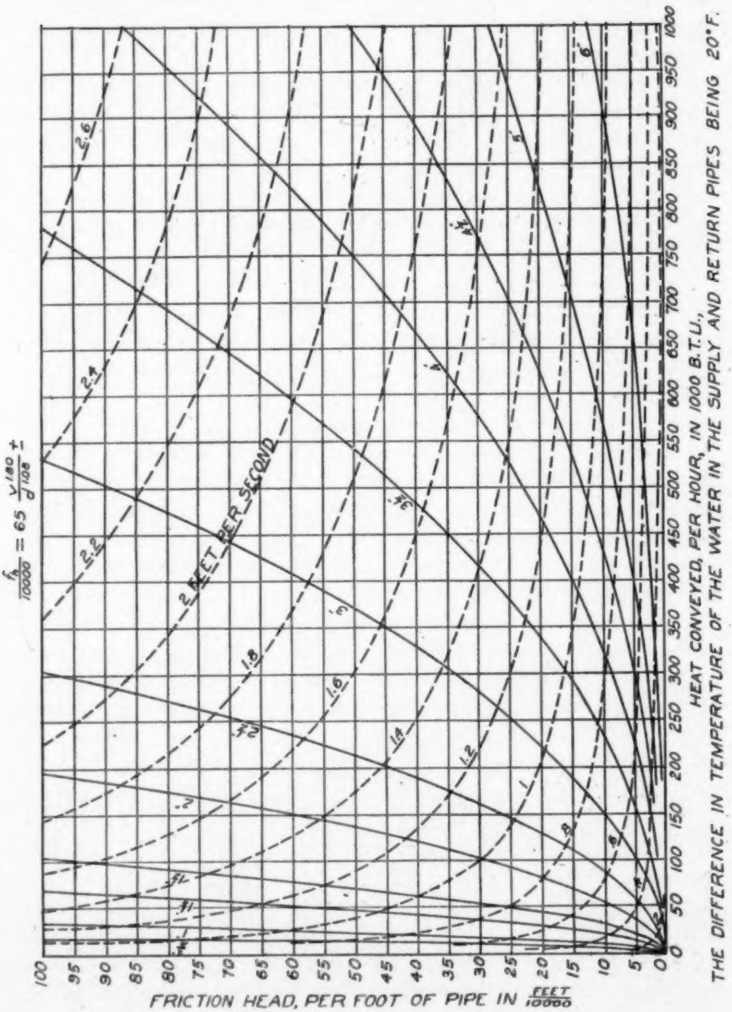


Fig. 8

laborious and it is highly desirable to simplify the work by the use of tables or diagrams.

Professor Rietschel prepared and published very complete tables showing the heat conveyed by the various standard German pipes for a large number of friction-heads and velocities and for several differences in the temperatures of the water; the writer has found it more convenient and sufficiently accurate to use diagrams similar to that shown in Fig. 8, which was calculated by means of the friction factors shown in Fig. 6.

To explain the use of the diagram of Fig. 8, let it be used to solve the problem stated above:—To convey 250,000 B.t.u. through a pipe 150 feet long by means of a pressure-head of .1415 feet when the difference in temperature of the water is 20 degrees.

A pressure-head of .1415 feet for a circuit 150 feet long is equivalent to .00094 feet per foot of pipe, or about 9½ feet in 10,000.

Find 250 in the lower margin of the diagram; follow the vertical line through that point and note that for a 3-inch pipe the friction is about 25 feet in 10,000, for a 3½-inch pipe about 11.4 and for a 4-inch pipe about 6.2; hence it follows that a 3-inch pipe is entirely too small, a 4-inch pipe is a little too large, and a 3½-inch pipe a little too small; a correct solution would require some 4-inch pipe and some 3½-inch; the respective lengths may be determined from the equation:

$11.4 \times x + 6.2\ (150 - x) = 1415$, $x = 93$ feet of 3½-inch pipe, and this means that there are required $150 - 93 = 57$ feet of 4-inch pipe.

The question arises: Is the diagram of Fig. 8 reliable? When it was constructed, the formulas employed were based on the friction factors published in works on hydraulics which represented the best available data at that time. Since then the German experiments have shown that as the temperature of the water increases the friction decreases; we may therefore conclude that the pipe sizes determined according to the diagram are unnecessarily large and that the diagram should be modified to conform to the results of the late German experiments.

It is difficult to prepare accurate diagrams for American pipe from the results of the German experiments because American pipe differs somewhat from the standard German pipe; in German work two distinct types of pipe are used, one known as Muffenrohre and the other as Siederohre.

Muffenrohre are similar to the American pipe with screwed couplings and are used in the smaller sizes only; Siederohre are

patented pipes with flanged couplings and are used in the larger sizes only. From the results of experiments, Professor Brabbee developed the following formulas for the two types of pipes, which represent the experimental results accurately within 4 per cent.

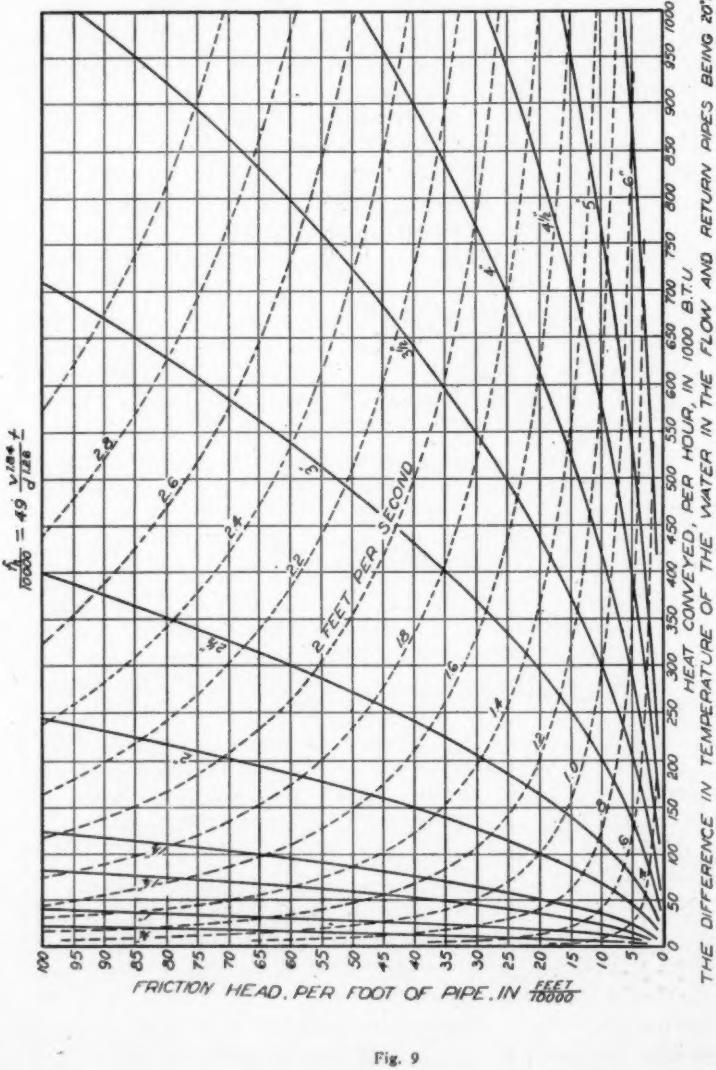


Fig. 9

For Muffenrohre $$\frac{h_f}{e} = 2570 \frac{v^{1.84}}{d^{1.26}}$$

For Siederohre $$\frac{h_f}{e} = 4920 \frac{v^{1.80}}{d^{1.37}}$$

In preparing the tables for Prof. Rietschel's book the Muffenrohre formula was used for pipes smaller than 2 inches and the Siederohre formula for pipes larger than 2 inches.

Translated into English units and expressed in feet per ten thousand, the two formulas become:

$$\frac{h_f}{10000} = 49 \frac{v^{1.84}}{d^{1.26}}$$

$$\frac{h_f}{10000} = 69 \frac{v^{1.80}}{d^{1.37}}$$

The curves of the diagram of Fig. 8 are expressed quite accurately by the formula:

$$\frac{h_f}{10000} = 65 \frac{v^{1.80}}{d^{1.08}}$$

To compare the two German formulas with each other and with the one just quoted the diagrams of Fig. 9 and Fig. 10 were designed to represent the two German formulas; they may be used as explained for Fig. 8.

An inspection of the three diagrams will show that for the problem stated above the friction head would be for 3-inch, 3½-inch, and 4-inch pipe respectively.

According to Fig. 9:—15.0, 7.2, and 4.0 feet/10,000.
According to Fig. 10:—18.4, 8.9, and 4.9 feet/10,000.
According to Fig. 8:—25.0, 11.4, and 6.2 feet/10,000.

The corresponding pipe sizes would be respectively:

According to Fig. 9:—43 feet of 3-inch and 107 feet of 3½-inch.
According to Fig. 10:— 8 feet of 3-inch and 142 feet of 3½-inch.
According to Fig. 8:—93 feet of 3½-inch and 57 feet of 4-inch.

This calculation shows that a considerable economy may be effected by designing heating systems according to coefficients based on the German experiments, but it also shows the desirability of conducting a series of accurate tests with standard American pipe.

If the pipe circuit is to be designed for a system in which the difference in the temperatures of the water is to be other than 20 degrees the same diagram may be used by varying the heat to be conveyed in the inverse ratio in which the difference in temperature varies from 20 degrees. For example, if the difference in temperatures, in the problem above, is to be 30 degrees instead of 20

degrees, the heat conveyed would be 30/20 as large as it would be for a difference in temperatures of 20 degrees, for the same velocity of flow, and consequently for the same friction head; hence we find 20/30 of 250, or 167, in the lower margin of the diagram and determine the friction-heads for the various pipe sizes and select

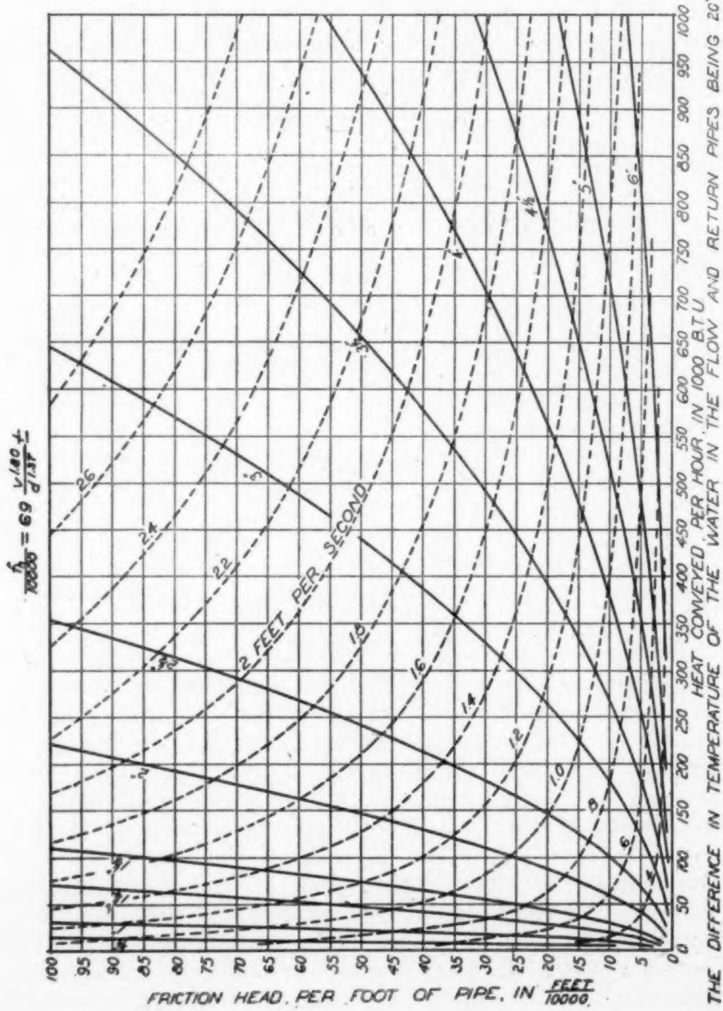


Fig. 10

the correct sizes, bearing in mind that the available pressure-head will be increased from .1415 feet to .2106 feet, according to Table 1. Thus, according to the diagram of Fig. 10, 48 feet of 2½-inch and 102 feet of 3-inch pipe would be required.

In the problem solved above the friction due to the pipe alone was considered.

In nearly all heating systems the friction due to the pipe, valves, radiators, etc., forms an important part of the total friction and must never be omitted.

The correct design of a heating system requires that the friction due to the pipe and the friction due to the fittings, valves, radiator, etc., for any one circuit, shall, together, be equal to the pressure-head for that circuit.

Prof. Rietschel represents the friction or resistance due to fittings, etc., by the expression, $s \times \frac{v^2}{2g}$, and assigns various values to s, which are based largely on experimental determinations. For example, if v and g are expressed in metric units, the values of s are as follows:—For 90 degree elbows from 1.5 to .5 and for tees from 2 to 1, depending on the size of the pipe; for radiators from 2½ to 3, and for various valves from 2 to 30. Tables are prepared, showing the friction-heads for various values of s and v, from which the friction-head due to any number of fittings may be determined and added to the friction-head due to the pipe.

By calculating the friction-head due to one elbow and that due to one foot of pipe, for the same velocity, the relation between the two can be determined, and the elbow expressed in terms of feet of pipe. This was done and the diagram of Fig. 11 prepared; the full lines show the relation between the same, according to Prof. Rietschel's tables.

To explain the use of the diagram, let it be required to find the friction due to four 3-inch elbows, when the velocity of the water is 1 foot per second. The diagram shows that for a 3-inch pipe and a velocity of 1 foot per second, one elbow is equivalent to 5.6 feet of 3-inch pipe; four elbows are equivalent to 22.4 feet; according to the diagram of Fig. 10, the friction head for a 3-inch pipe and a velocity of 1 foot per second, is 15 feet in 10,000, or .336 feet for 22.4 feet; this is also the friction-head due to the four elbows.

The writer conducted a series of experiments during the spring and summer of 1912 to determine the friction in elbows and other fittings and published the principal results in Domestic Engineering Nov. 2, 1912; in these tests eight elbows were tested at one time

so that in the final results all errors and inaccuracies are divided by eight.

Fig. 12 shows the results of these tests on 1-inch galvanized iron

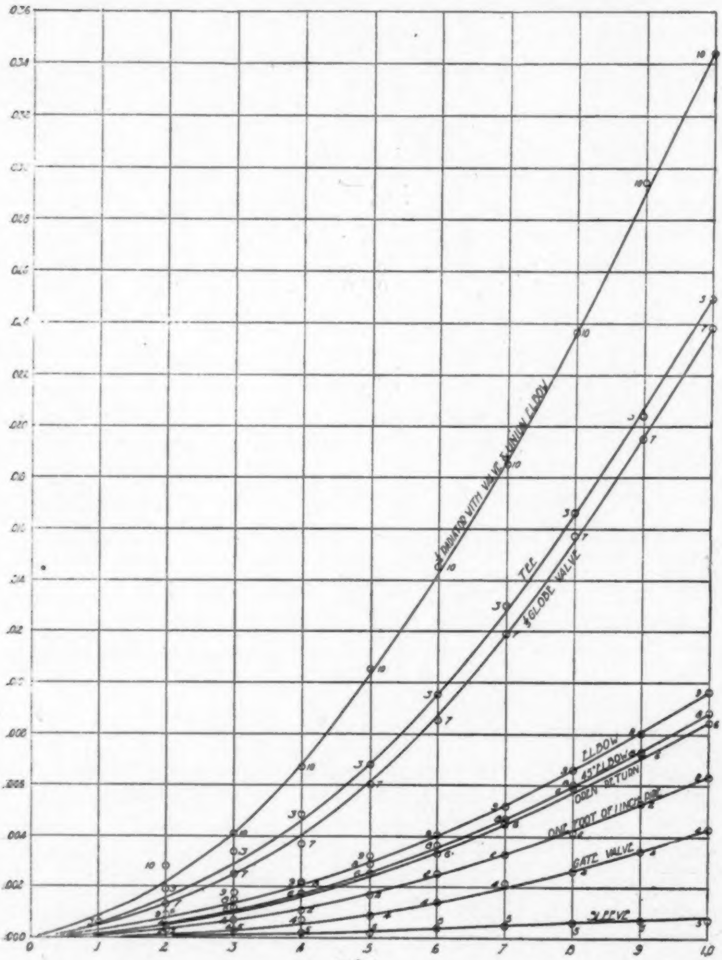


FIG. 12

pipe, sleeves, gate valves, open return bends, 45 degree elbows, 90 degree elbows, globe valves, tees, and radiators with radiator valves and union elbows.

From this and other tests the writer concluded that the following relations may be assumed between the ordinary 90 degree elbow and other fittings, valves, etc.:

1 45 degree elbow equivalent to half of a 90 degree elbow.

1 open return bend equivalent to one 90 degree elbow.

1 gate valve equivalent to half of a 90 degree elbow.

1 globe valve equivalent to twelve 90 degree elbows.

1 tee equivalent to two 90 degree elbows.

1 sleeve (negligible) equivalent to one-tenth of a 90 degree elbow.

1 radiator, with valve and union elbow, equivalent to seven 90 degree elbows.

Having found the number of 90 degree elbows equivalent to the various fittings used in any circuit, their equivalent length of pipe may be determined from Fig. 11, this added to the actual length of pipe in the circuit, and for this sum the friction-head determined as explained above.

The curves of the diagram of Fig. 11 do not agree closely with the result of the writer's tests of elbows; to show the differences

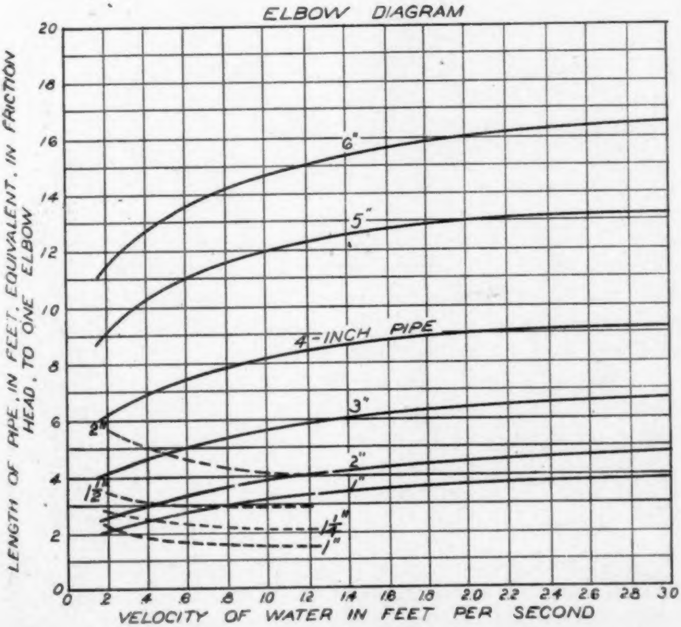


Fig. 11

between the two, the dotted lines of Fig. 11 were drawn; they represent the results obtained by the writer for the relations between elbow and pipe, for velocities ranging from .2 to 1.2 feet per second, and for pipes ranging from 1-inch to 2-inch.

The differences between the dotted and the full lines of Fig. 11 suggest the desirability of additional tests to determine the friction due to elbows, etc.

ABSTRACT OF THE PAPER

This paper deals largely with the German practice in hot water heating, and the work of Prof. Rietschel along these lines is explained and contrasted by charts with the older work of Weisbach.

Tables are given showing the pressure head necessary for the supply of a given radiator or system of radiators and sketches are shown illustrating the various pressure heads at different levels.

The paper gives charts showing the Rietschel friction factors contrasted also with the older ones of Weisbach.

The paper also gives explanations as to the friction in valves and fittings of various kinds and the effect they have on the water flowing in the pipes in a hot water heating system.

The article gives a sketch representing the piping system in the Academic Building of the University of Texas in which there is a water heating system laid out according to the Reitschel method.

DISCUSSION

Arthur K. Ohmes: Prof. Giesecke is entitled to the grateful thanks of the members of the Society for the excellent paper as presented. There is nothing that I can add to the purpose of the paper, which was to describe the method of Prof. Rietschel for determining pipe sizes for hot water heating systems. As to the incidental remarks of Prof. Giesecke on friction coefficients, resistance of elbows, valves, etc., may I be permitted to say the following:

While Prof. Rietschel's method (as evolved by him) is universally used in Germany, still the friction coefficient of Weisbach, $f = .014 + \frac{.0172}{\sqrt{v}}$ which Rietschel adopted, has not been used for many years by some of the best concerns in their private practice. It has been known in Germany for many years that

the old Weisbach friction coefficient is too high at ordinary velocities we meet with in gravity hot water heating systems.

In using the Weisbach friction coefficient we erred therefore in most cases on the safe side. As I remember it, Prof. Rietschel himself at a time pointed this out, but he also stated that there was no objection to this, as long as a certain coefficient is consistently applied to all pipe sizes. It simply means that the gravity apparatus would ordinarily circulate with a smaller temperature difference and in forced hot water with a smaller pump head, than assumed in the calculations. Nevertheless the keen competition caused the heating engineers to look in their private practice for friction constants which would come more nearly to the exact conditions.

It is my recollection that many concerns used the friction coefficients of Lang. Lang advised after hundreds of experiments the following formula for figuring friction coefficient:

$f = a + \frac{b}{\sqrt{vd}}$ in which f = frictional constant; $a = .02$; $b = .00328$.

v = velocity of water in feet per second.

d = diameter of pipe in feet.

If the frictional constants are determined from this formula they will come half way between those of figures 6 and 7.

Lang's coefficients were used for hot and cold water, and though more correct were still too high for hot water heating systems. It is a well-known fact that on account of the viscosity of water at a high temperature, as we deal with it in hot water heating systems when the maximum amount of water must be circulated, less head is required to circulate a certain quantity of water than if the water were cold. The entire question of the head required for moving liquids and gases in confined spaces was most thoroughly investigated for the Society of German Engineers by Dr. Biel, who most beautifully described in the Journal of the Society of German Engineers, 1908, page 1035.

In this classic description of the friction losses of moving fluids Dr. Biel advises in the main three formulas:

1st. Parallel motion.

2nd. Partly parallel and partly turbulent motion.

3rd. Turbulent motion.

These formulae are to be applied with additional factors and constants for varying viscosity at low and high temperature of the fluid and for the roughness of the pipes themselves. Apply-

ing Dr. Biel's formulae to hot water heating apparatus we find that the pipes again became somewhat smaller than by the Lang coefficient and no doubt more accurate. Dr. Biel's formulae were utilized in the tables published by Recknagel and these tables are yet most widely used in Germany.

Not being satisfied with the already extensive researches made in this important field, it was again by means of the latest testing instruments, etc., gone over by Dr. Brabbee, as Prof. Giesecke mentions. Some of these results are shown in Figure 7. A very complete analysis of these tests can be found in "Heft 5 der Mitteilungen der Prüfungsanstalt für Heizungs- und Lüftungs-Einrichtungen der Kgl. Technischen Hochschule zu Berlin.

Determining pipes according to Dr. Brabbee's formula will still make them smaller than by Dr. Biel's formula. Dr. Brabbee gives in addition to the two formulas quoted by Prof. Giesecke (which are for hot water) also similar formulas for cold water As was pointed out in the Gesundheits-Jugemein, page 568, year 1914, that if all four formulas are applied to pipes above 40 inches diameter a larger loss would be determined for a given length of pipe for moving cold water than for moving hot water. This clearly cannot be right.

It is questionable therefore that even Dr. Brabbee's formulas will stand a test of time, and caution is advised in applying them without careful comparison to our American pipe sizes, fittings, valves, etc. On the other hand, I do not wish to question the fact that any and all of the formulas will not produce a good working apparatus. The only difference may be that the apparatus will circulate at a lesser or high difference of temperature or that more or less pump head is required in the case of forced hot water, than that which was assumed in the calculations. In both cases I believe the best economy can be secured when the pipes are fairly ample.

After all, a hot water heating system is the only one which has a self-adjusting circulating feature. By this I mean that if a radiator connection is slightly too small the amount of water going into an otherwise properly porportioned radiator will be less than assumed. This in turn means that the water in this particular radiator is cooled more than figured as well as in an adjoining radiator receiving its proper quota of water.

A lower temperature of water means that the circulating head is increased and with it a more positive circulation making up to a great extent for the loss of the smaller pipe.

After all, under equal pressure losses the carrying capacities of most of our commercial pipes vary nearly 100 per cent. It is evident that even in the best designed pipe systems there must be cases like the one above described.

Except for the self adjusting feature of the circulation in our hot water heating systems there would be many sad experiences in securing the proper circulation even in our most carefully calculated hot water heating plants.

From the same Heft 5 der Mitteilungen, etc.

We also find that the pipe frictional resistances do not vary with the square of the velocity, whereas the resistance of elbows and valves, diverging and converging water currents in T-fittings, are varying in direct proportion to the square of the velocity. It is therefore possible to express head of elbows, etc., directly in a proportion to the theoretical velocity head $h = \frac{v^2}{2g}$.

By inserting the proper density and other factors the losses of elbows, valves, T's, etc., can be expressed in a measurable height of velocity head. Much time can be saved by preparing tables showing values of velocity heads under varying velocities, because it is simpler to add these to the losses occasioned by the friction of the water in the straight pipes than by converting the loss of head occasioned by a fitting into an equal pipe length and then figure the frictional losses in the regular way.

CCCLXXXV

APPARATUS FOR THE STUDY OF HEAT RADIATION

J. D. HOFFMAN.

In the mass of material that collects in the reference files of our members there is no doubt many an idea lying dormant, which if submitted to the society for open discussion would be of considerable value. Since the American Society of Heating and Ventilating Engineers is a recognized clearing house for special scientific investigation, taking in accumulated ideas and data and giving out digested facts, I am proposing several questions relating to the combustion of fuels for the society's consideration. First, in burning a stated amount of fuel does the total amount of heat given off as radiant energy remain the same for all rates of combustion? That is, having a number of equal samples of coal to be completely burned, each sample to vary in time of burning, will the heat radiated be a constant amount? Second, is there a definite relation between the heat given off as radiant energy and that carried off by convection? Third, is there one rate of combustion for every fuel that is the most economical rate? Fourth, in burning a given amount of fuel in a given time is it more economical to reduce the grate area and increase the rate of combustion?

It is not the purpose of this brief paper to attempt to answer any of these questions. They are thrown in merely to stimulate discussion in the society journal or at the society meetings. Feeling an interest in such an investigation, I outlined a piece of apparatus which was developed by Messrs. G. C. Polk and L. C. Lichty, two senior students in the Mechanical Engineering Department of the University of Nebraska who, after finishing the apparatus, conducted a number of tests upon it. The data obtained seemed very encouraging but not at all conclusive. It is evident that much more work will have to be done if dependable results are to be obtained. A description of the apparatus may be of interest to the members.

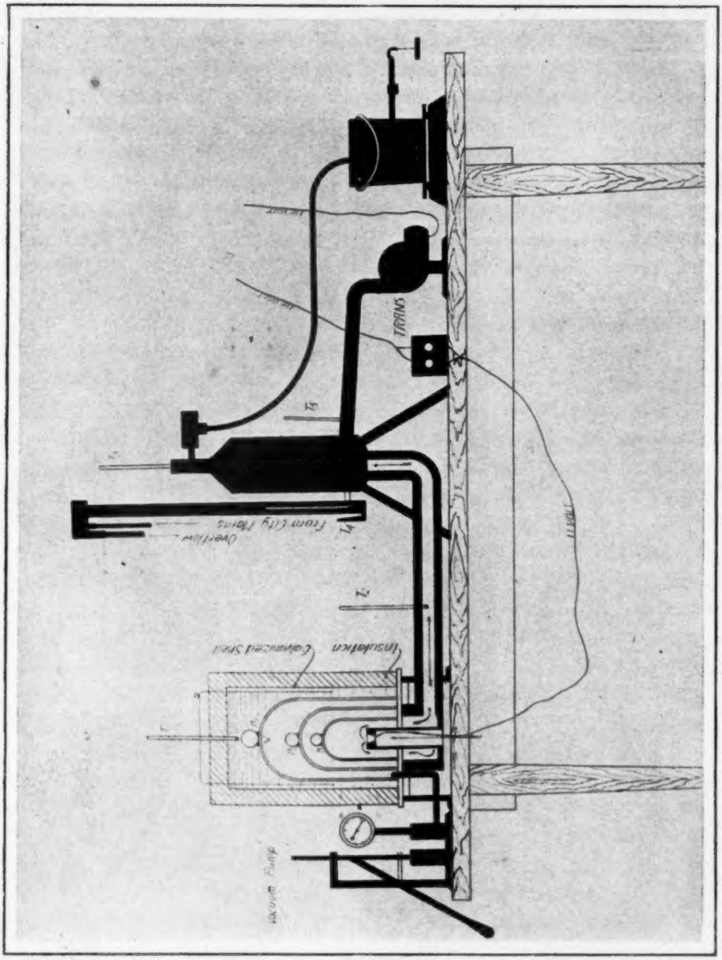


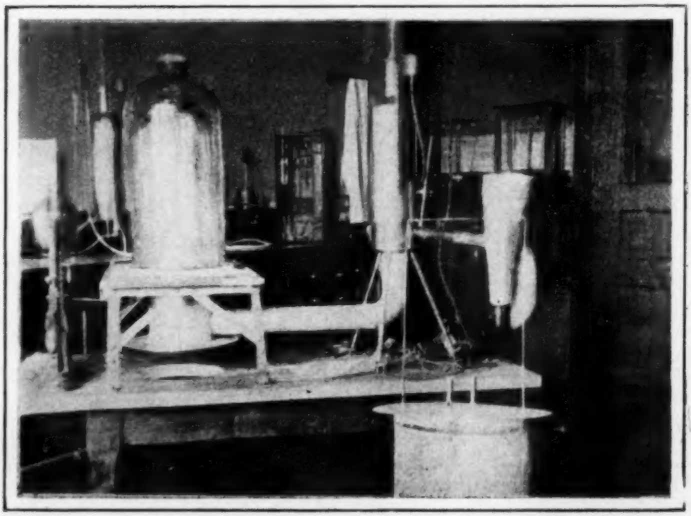
Fig. 1

Fig. 1 shows the assembled apparatus. It comprises two calorimeters (one to measure the heat radiated and the other to measure the convected heat), a vacuum pump, a transformer, an electric motor-fan and a weighing scale. All the apparatus is standard excepting the calorimeter for measuring the heat radiated. This consists of a metal fire tube enclosing a clay fire pot through which the room air enters the combustion chamber (this tube is adjustable

for height); one glass bell-jar, B_1, to protect the bell-jar B_2 from excessive heat; two bell-jars B_2 and B_3 enclosing an air tight compartment V (this compartment is connected with the vacuum pump); and a galvanized tank heavily insulated, forming a water chamber around bell-jar B_3. This water chamber is provided with a stirring device which is used just before taking readings. A peep-hole is arranged through the insulation at the side of the calorimeter and a reflector below the fire tube to observe the fire. Provision is made for temperature readings as follows:

Temperature of room.............................T
Temperature of water in radiant heat calorimeter.....T_1
Temperature of gases leaving fire chamber...........T_2
Temperature of water leaving Junker calorimeter.....T_3
Temperature of water entering Junker calorimeter....T_4
Temperature of gases leaving Junker calorimeter.....T_5

The desired velocity and volume of the gases are obtained by a variable speed motor-fan and an anemometer located at the fan outlet. The pipe carrying the gases between the two calorimeters is thoroughly insulated. Any air pressure may be maintained in chamber V. During the series of tests four degrees of pressure were maintained; atmospheric, ten inches, twenty inches and twenty-eight inches. To assist the fire in starting, an eleven volt circuit with platinum glow is placed in the fire pot. The equivalent heat



given off by this circuit is finally deducted from the experimental summation for the heat balance. Fig. 2 shows the apparatus in the process of construction and Fig. 3 shows it completed.

The *method of procedure* in burning any one sample of fuel is as follows: water is circulated through the Junker calorimeter and upon the temperatures T_3 and T_4 becoming constant the switch is closed thus lighting the fire and starting the motor. The fuel usually begins to flame in one minute and the wire is kept hot for two minutes. After the first two minutes the speed of the air is

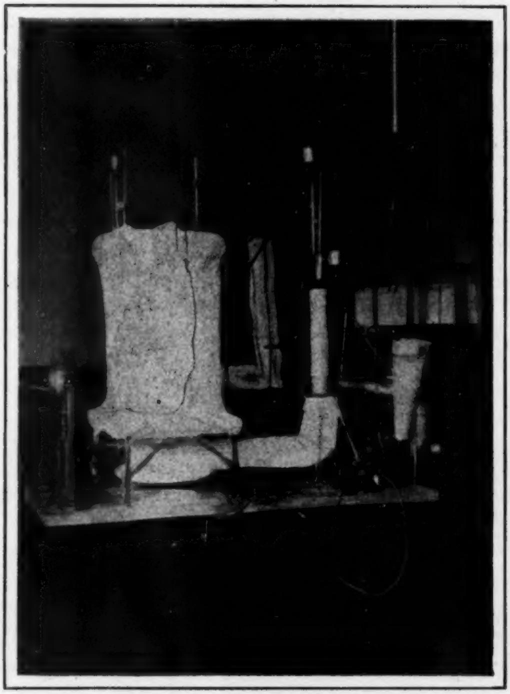


Fig. 3

regulated to some definite speed so as to give a uniform rate of burning. As soon as the fire is out the velocity of the air through the anemometer is increased to one hundred feet per minute and kept constant until the end of the test to cool down the apparatus. All tests are run until the temperature of the incoming gases T_2 becomes constant, which usually occurs at one or two degrees above the room temperature. Then the fan is shut down and the water

allowed to keep flowing through the Junker calorimeter until the difference between T_3 and T_4 is the same as at the beginning of the test. All thermometer and anemometer readings are taken every two minutes. The water is weighed at convenient times during the test. The residue left in the fire pot is weighed after each test and deducted from the amount in the original sample in each case. Analysis of heat values may be made of some of the original samples to obtain a basis of comparison.

The improvised calorimeter has given very satisfactory results in all points but two. First, the sample of fuel has never completely burned. Fire pots of varying shapes and sizes were tried and the total heat obtained from the test, i.e., the heat radiated across the vacuum to the water, plus that obtained in the Junker calorimeter, minus the heat equivalent of the electric circuit, never exceeded ninety-five per cent. of the heat value of the fuel as obtained by separate tests. Second, fuels with high hydrocarbon content had a tendency to coat the inner bell-jar and retard the radiant ray. No difficulty was experienced in maintaining the highest degree of vacuum in the space V or in otherwise handling the apparatus.

All things considered, the experimental results thus far obtained seem to justify further research and we hope this may be done in the near future.

DISCUSSION

William Kent: The apparatus described by Prof. Hoffman is interesting as an attempt to solve some of the obscure problems of heat radiation.

The four questions may be answered from our present knowledge of heat phenomena without any new experimental research. It is generally accepted that the amount of heat radiated from a hot body into a cooler one depends on the nature and on the color of their surfaces, and is also some logarithmic function of the temperature of the two bodies. Stefan and Boltzman's law given in books on physics, states that the energy radiated by a "black body" is proportioned to the fourth power of the absolute temperature or $E = (T^4 - To^4)$ where K is an experimental constant, T the temperature of the hotter body and To that of the cooler, also that the total radiation from other than black bodies increases more rapidly than the fourth power of the absolute temperature, but no one has yet shown any application of this law to the solution of any practical problem in heating and

ventilation. Possibly Prof. Hoffman's apparatus may be used to determine whether the Stefan and Boltzman law is true or not, and to determine values, the constant K under different conditions other than those of temperature.

Whether the law is true or not, there can be no doubt that the radiant energy transmitted increases very rapidly with increase of temperature, and therefore if the answer to Hoffman's first question "In burning a stated amount of fuel does the total amount of heat given off as radiant energy remain the same for all rates of combustion," depends upon the answer to another question, "In burning a stated amount of fuel does the temperature remain the same for all rates of combustion?" The answer to this second question as well as to the first is "No," for the loss of heat by radiation expressed by a percentage of the total heating value of the fuel burned, increases as the rate of combustion is diminished, since the total radiation loss increases with the time it takes to burn the fuel.

If there were no radiation loss, the furnace being perfectly insulated, the temperature would be independent of the time, but if there is a loss by radiation then the amount of heat available for increasing the temperature would be less than the total heating value of the fuel, and the temperature of the furnace on which the radiation loss depends, would be diminished. There are thus two influences resulting from decreasing the rate of combustion, which attain opposite directions. The first is an increased radiation loss due to the longer time in which the radiation takes place, and a consequent decrease of temperature, and the second is a decreased radiation loss due to this increase of temperature, and it would be only an accidental case in which these two influences would balance each other. The question may also be answered "No" on other grounds. The temperature of the furnace depends not only on the radiation loss, but also on the amount of excess air used to burn the fuel. It is possible to have a temperature of over 3,000 degrees Fahr. in the furnace if it is well insulated and the air supply is regulated so that the flue gases contain only from 3 to 5 per cent. of free oxygen, but the temperature may be as low as 1,500 degrees Fahr. if the air supply is greatly in excess, so that the gases contain say 15 per cent. of oxygen.

The answer to Prof. Hoffman's second question, "Is there a definite relation between the heat given off as radiant energy and that carried off by convection?" also is "No," for the rea-

son that the heat carried off by convection depends on the character of the surface, on the surface, or the temperature of the air or other fluid surrounding the surface and on the velocity. There can be no definite relation between this quantity depending on so many variables and another quantity which depends on an entirely different set of variables.

To. Prof. Hoffman's third question "Is there one rate of combustion for every given fuel that is the most economical rate?" the answer is "No." Neglecting the loss due to radiation, which in factory boilers may be reduced to as low as 1 or 2 per cent. of the total heating value of the fuel, the temperature of the furnace with equal conditions as to the air supply is independent of the time. The economy of the fuel is theoretically independent of the rate of combustion, and practically as good economy has been obtained in steam boiler practice with a rate of combustion of only 6 pounds of coal per square foot of grate per hour as with 20, 30 or 40 pounds provided in the case of slow driving the air supply conditions are properly controlled, and in the case of the most rapid driving shaking grates or other devices are used to keep the fire bed free from ash and clinker.

The answer to the fourth question "Is it more economical to reduce the grate area and increase the rate of combustion?" is theoretically "No," practically it is often less economical, for rapid combustion tends to increase ash and clinker troubles and prevent proper regulation of the air supply.

If further experiments are to be undertaken with Prof. Hoffman's apparatus I suggest that it be modified in two particulars, first, by replacing the fuel furnace by an electric furnace, the amount of heat energy entering the apparatus being determined by means of a watt meter, and second, by replacing the anemometer by some more accurate means of measuring the air.

CCCLXXXVI.

CAN WE LOCATE THE NEUTRAL ZONE IN HEATED BUILDINGS?*

J. J. BLACKMORE

A study of the conditions created by the application of artificial heat to our buildings presents some very interesting and fascinating problems. Some of these problems are very elusive and not easily solved.

The conditions thus created, however, are due to the working of physical laws and if these laws are studied and understood, the reason for the changed conditions is readily appreciated.

The location of the neutral zone is one of these problems and its location is important on account of the many disturbances which may be created in the design and operation of heating and ventilating plants; especially is this the case in a large building of many stories in height.

In cold weather, the air contained in a heated building, being of a higher temperature than the outside, is much lighter than the air surrounding it, and, being lighter, it has a tendency to rise in accordance with the laws of gravitation. This body of light air is kept from rising by the walls and roof of the building. being thus prevented from rising, pressure is exerted against the ceiling and upper part of the walls to a considerable extent. In a large building say 200 feet by 200 feet by 125 feet containing 5,000,000 cubic feet, the difference between the weight of the inside air at 70 degrees, and the same quantity outside at zero is $5{,}000{,}000 \times (.0864 - .075) = 57{,}000$ lbs. or 28.5 tons, .0864 being the weight in pounds of a cubic foot of dry air at zero temperature and .075 being the weight in pounds of a cubic foot of dry air at 70 degrees temperature.

As an example, we will consider the conditions in a room as shown in Fig. 1, which is supposed to be air tight and uniformly

* The data in this paper formed the substance of a lecture in a course on heating and ventilation given by Mr. Arthur K. Ohmes to a class of Columbia University students.

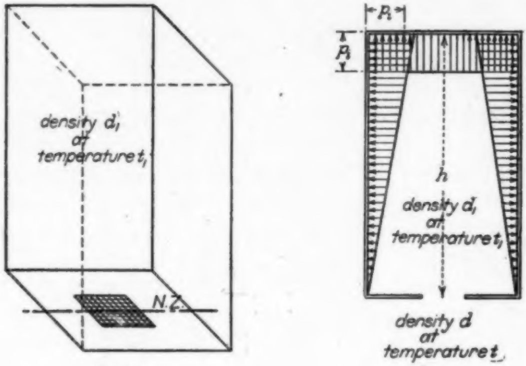


Fig. 1

heated to 70 deg. when the outside air is at zero. The room having an opening at bottom the atmospheric pressure would be equalized at this opening.

The weight of a column of air one foot square and fifty feet high at 70 deg., would be $50 \times .075 = 3.75$ lbs., and a similar column of air at zero would weigh $50 \times .0864 = 4.32$. The difference being .57 lbs., per square foot.

The warm air is, therefore, pressed towards the ceiling by the greater density of the external air pressure exerted through the opening in the bottom of the room, the difference in weight or density of the two bodies is manifested by pressure against the ceiling and walls of the room. This difference of pressure would create a velocity equal to approximately twenty-two feet per second, at the above temperatures. $V = \sqrt{\frac{2\,G\,P}{Y}}$ or $V = \sqrt{\frac{2 \times 32.16 \times .57}{.075}} = 22$. In which G is the acceleration due to gravity, P the pressure in pounds per sq. foot, and Y the weight of a cubic foot of air at 70 deg. in pounds.

Comparison may be made between conditions in such a room and an imaginary U tube of large dimensions as shown in Fig. 2. Assuming that the U tube is fifty feet high, filled on one side with water and the upper end of the tube on that side is closed tight. Into the other side is introduced the same volume of mercury. The column of water fifty feet high would exert a pressure of about 21 lbs., at the base of the tube, while the mercury would exert a pressure of 273 lbs., at the same point. In

this case the pressure due to the mercury would force the water up against the closed end of the tube equal to a pressure of 273 — 21 = 252 lbs.

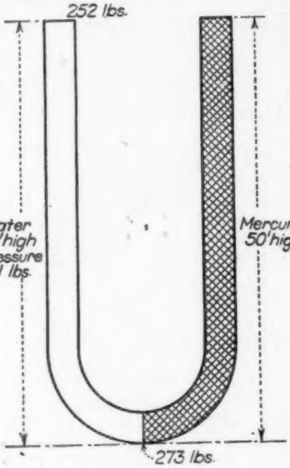


Fig. 2

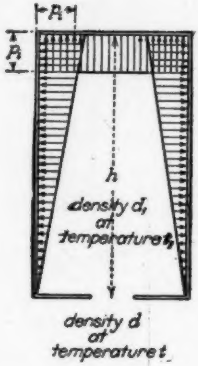


Fig. 3

The different pressures in a room like that shown in Fig. 1 may be represented by the illustration Fig. 3, and the pressure at the top of this room as shown by the example of the U tube would be Pi = h $(d - d_1)$ in which

Pi = pressure above the atmosphere;
d = density of the outside air;
d_1 = density of inside air.

Having in mind that one cubic foot of air at zero weighs .0864 lbs. per cubic foot and that for other temperatures the density of the air is in direct proportion to the absolute temperature, we can express the equation with approximate accuracy in temperatures to obtain the difference in density between equal volumes of the inside and outside air, as follows; for a room fifty feet high.

$$Pi = h \quad \left(\frac{.0864 \times 460}{460 + t}\right) - \left(\frac{.0864 \times 460}{460 + t_1}\right)$$

or $Pi = 50\left(\frac{.0864 \times 460}{460 + 0}\right) - \left(\frac{.0864 \times 460}{460 + 70}\right) =$ .57 lbs. per square foot

If this same room with the same conditions as before had one opening at the top instead of the bottom as in Fig. 4, and if

the atmospheric pressure equalized with the room pressure, at this point, the difference in weight of the two columns of air would be the same as in the case with the opening at the bottom, but the external air being heavier would press on the under side of the room with greater force because the outside air is of greater density than that within. If an opening were made in the bottom, the room would appear to be under a vacuum for the heavier air would at once flow through the opening into the room.

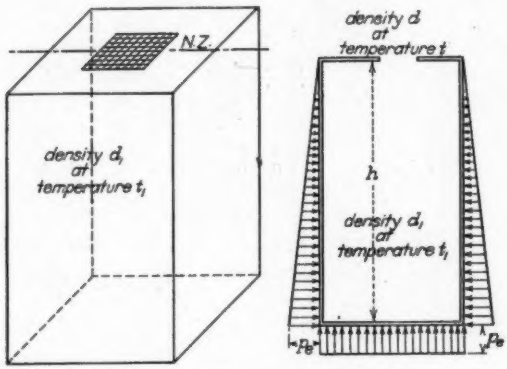


Fig. 4

We have designated the pressure exerted against the inside walls as Pi, and in a similar manner we will designate the pressure against the outer surface of the walls as Pe, then the equation to obtain the difference in weight or pressure by the density is:

$Pe = h\ (d - d_1)$, or to obtain it from the temperature it will be:

$$Pe = h\left(\frac{.0864 \times 460}{460 + t}\right) - \left(\frac{.0864 \times 460}{460 + t_1}\right)$$

The conditions illustrated in these two extreme examples show clearly the fact that different pressures above or below the outside pressures must exist in a room or building when the contents are heated to a temperature above that of the external air.

In a similar manner if a room or building is cooled in the summer months the reverse of these conditions will obtain.

Additional openings into rooms will produce different conditions, also a change in the location of the openings into the room will shift the location of the neutral zone as will be shown in the following examples.

In Fig. 5, with an opening in the side of the room the neutral zone will be slightly above the center of the opening and the following formula will apply:

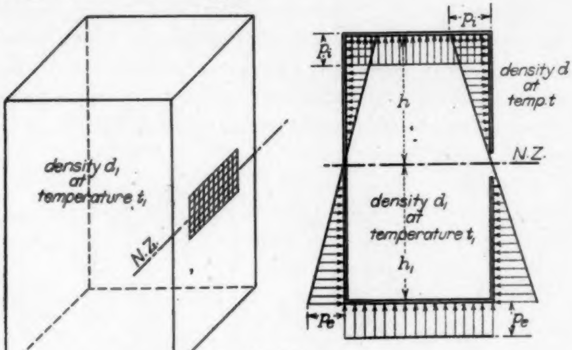


Fig. 5

$Pi = h_1 + h\ (d - d_1)$ to obtain the pressure by density.

$$Pi = h_1 + h\left(\frac{.0864 \times 460}{460 + t}\right) - \left(\frac{.0864 \times 460}{460 + t_1}\right) \text{to obtain pressure}$$

by temperature.

$Pe = h_1 + h\ (d - d_1)$ to obtain the pressure by density.

$$Pe = h_1 + h\left(\frac{.0864 \times 460}{460 + t}\right) - \left(\frac{.0864 \times 460}{460 + t_1}\right) \text{to obtain pressure}$$

by temperature.

Fig. 6, illustrates the location of the neutral zone as it would be in the case of toilets, kitchens, or other rooms generating odors

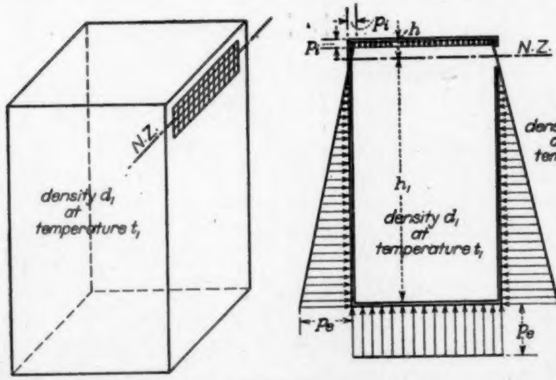


Fig. 6

which necessitate air opening for ventilation near the ceiling. In such a room a strong draft will be noticed when doors are opened into it. The same formula applies as given for Fig. 5.

Fig. 7 illustrates the location of the zone in a room with an opening near the floor as would be the case if a room was heated by the indirect method, if the register was located in a similar place. The opening of a door into such a room would occasion little if any draft. The same formula applies as given for Fig. 5.

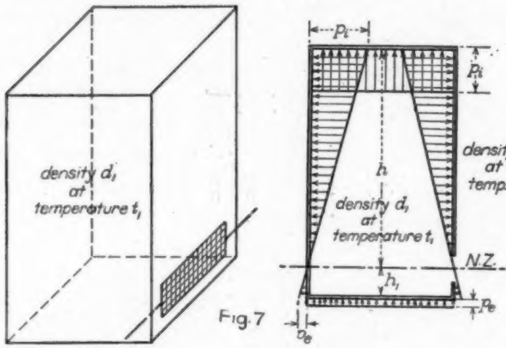


Fig. 7

Fig. 8 illustrates a room with an inlet and outlet which equalizes the conditions and brings the zone to a point just above the center. Of course an accelerated draft in the upper opening would raise the zone to a higher plane and a partial closing of the same opening would lower it to balance the changed condition. The same formula applies as given for Fig. 5.

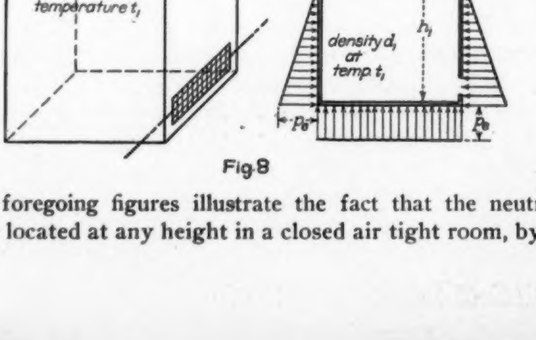


Fig. 8

The foregoing figures illustrate the fact that the neutral zone may be located at any height in a closed air tight room, by chang-

ing the position of an opening to the outer air. It is also possible to locate the neutral zone at a distance above the ceiling or below the floor of a room by an arrangement of outlets as shown in Figs. 9 and 10.

Fig. 9 illustrates a means of locating the zone at a point below the floor line. A floor register and a ventilating pipe with a strong draft would produce this condition:

$Pi = h_1 + h\ (d - d_1)$ to obtain the pressure by density.

$Pi = h_1 + h \left(\frac{.0864 \times 460}{460 + t}\right) - \left(\frac{.0864 \times 460}{460 + t_1}\right)$ to obtain the pressure by temperature, being the pressure at ceiling line inside.

$Pi_1 = h_1 + h\ (d - d_1)$ to obtain the pressure by density.

$Pi_1 = h_1 + h \left(\frac{.0864 \times 460}{460 + t}\right) - \left(\frac{.0864 \times 460}{460 + t_1}\right)$ to obtain the pressure by temperature, being the pressure at floor line inside.

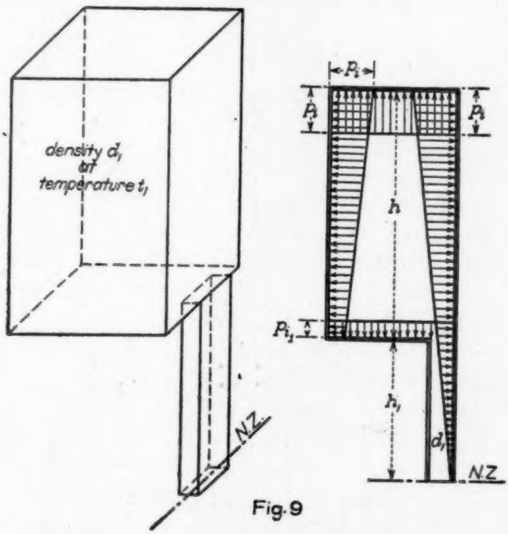


Fig. 9

Fig. 10 illustrates a condition the reverse of those in Fig. 9, a ventilating pipe from the ceiling would lift the neutral zone to a point above the ceiling about as shown.

$Pe = h_1 + h\ (d - d_1)$ to obtain the pressure by density.

$Pe = h_1 + h\left(\frac{.0864 \times 460}{460 + t}\right) - \left(\frac{.0864 \times 460}{460 + t_1}\right)$ to obtain pressure by temperature, being the pressure acting from the outside on the floor line.

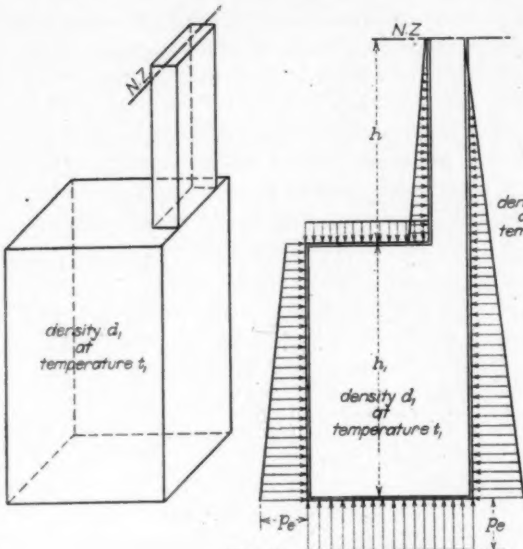


Fig. 10

$Pe_1 = h_1 + h\ (d - d_1)$ to obtain the pressure by density.

$Pe_1 = h_1 + h\left(\dfrac{.0864 \times 460}{460 + t}\right) - \left(\dfrac{.0864 \times 460}{460 + t_1}\right)$ to obtain pressure by temperature, being the pressure acting from the outside on the ceiling line.

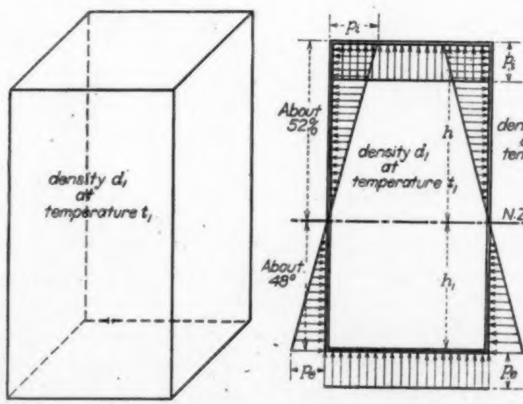


Fig. 11

In an air tight room as shown in Fig. 11 the neutral zone would be 2 per cent. below the center, that is, it would be located at about 48 per cent. above the floor line.

Again the author calls attention to the fact that all these examples are based on a condition of still air and tightly sealed enclosing surfaces, a condition which never exists in our buildings because the walls, ceilings and floors are porous and the windows and doors are surrounded by many cracks and crevices through which the air leaks in or is forced out as may be determined by their location in relation to the neutral zone.

This condition may be explained as follows: There is always a place or plane inside a room or building that will be equal in pressure to that of the air surrounding it on the outside. This plane has been rightly called the "neutral zone." Its exact location in a room or building depends on the relative leakages at the ceiling or upper part of the walls or at the floor and lower part of the walls and under ordinary conditions in still air its location will be slightly below the central horizontal plane of the room. It will be understood, however, that if the lower portion of a building has more openings through which air can leak in than the upper portion, the zone would be raised somewhat higher and the reverse would be the case if more openings were located in the upper part.

No doubt the location of the neutral zone of an entire building will vary greatly with its size and shape, with the number and size of the windows and doors and their relative location, etc.

We will now consider the effect on the neutral zone of the shape of some noted buildings in New York City which, for the purpose of illustration, we will assume are all forty stories in height. The neutral zone will prove to be at an approximately different level for each building, if the location of it is figured from the foregoing data.

The location of the neutral zone in a building of the same floor area all the way up, as shown in Fig. 12, will be about at the nineteenth floor. A building, as shown in Fig. 13, with a central tower above the twenty-fifth floor would have the neutral zone at a lower level, and in this case it would be somewhere about the sixteenth floor. A building, as shown in Fig. 14, with a tower above the eighteenth floor would have a still lower neutral zone, and in this case it would be approximately at the twelfth floor. In the case of the building illustrated in Fig. 15, which has a tower extending above the ninth floor, the neutral zone would be lowered

to somewhere about the ninth floor, or practically at the roof line of the main building.

The practical importance of these illustrations would seem to be that they show that a rectangular or square building would have the greatest indraft and the least pressure at the top, whereas the tower buildings would have a less indraft per foot of surface, but a much greater pressure at the top of the tower.

Anyone who has visited the top of a building like the Woolworth or Singer tower has probably noticed how difficult it is to close a door against the current of air flowing out from the top of such a building.

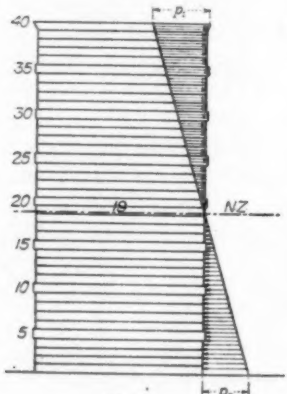


Fig. 12

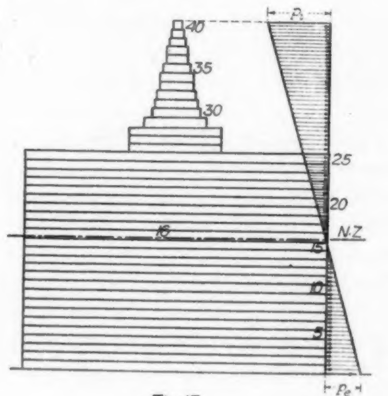


Fig. 13

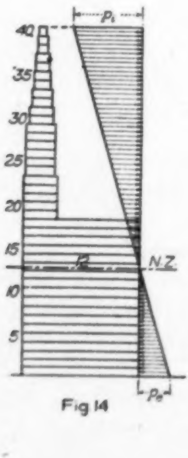


Fig 14

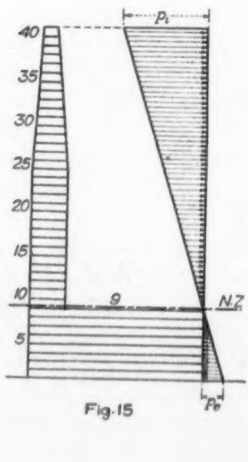


Fig. 15

If we consider a tower building seven hundred and fifty feet high and assume its neutral zone to be 300 feet above the base, we would have (with a temperature of zero outside and 70 deg. inside) an outside pressure of:

Pe = 300 (.0864 — .075) = 3.42 lbs. per sq. ft. or 55 feet per second, = 36 miles per hour and an inside pressure of:

Pe = 450 (.0864 — .075) = 5.13 lbs. per sq. ft. or 66 feet per second, = 45 miles per hour.

Any high winds will effect these velocities as follows:

On the lower floors the leakages would be increased but on the upper the pressure would be counteracted because the tower presents a smaller surface to the wind.

It is, of course, understood that these studies on the subject of the neutral zone are considered with the building located in still air, wind velocities will change the location of the neutral zone, but such changes of location will not be material.

Poor or leaky construction in a building will materially affect the heat required to offset the loss occasioned by such construction, but such defects will not change to any great extent the location of the neutral zone. If more air leaks in below the neutral zone, a correspondingly larger amount will be forced out above the neutral zone.

Wind at a high velocity blowing against one side of a building may cause air to leak in to a larger extent on that side of the building on account of the excess pressure, in which case more air is forced out from the sheltered sides of the building which leaves the neutral zone not materially changed from what it would be in still air, but a larger amount of radiation would be needed on the windy side of the building to maintain a uniform temperature on the inside.

It will be seen from the foregoing that the radiation provided for the space below the neutral zone will need to be much in excess of that provided for the space above the neutral zone with a possible exception in the case of a top floor which is affected by the heat losses from the roof.

In a similar manner the neutral zone affects the ventilating system of a building. Its location determines the pressures in the building, and the pressures must be added or subtracted to or from the pressures under which the fans operate, whether the fans are placed in the basement or on the top floor of the building.

This subject is so important that it is surprising that it has not been discussed in the transactions of the Society heretofore

and the author hopes this paper will be the means of starting a discussion that will continue till the neutral zone becomes a familiar acquaintance.

ABSTRACT OF PAPER

The paper deals with the problem of the difference in pressure between the inside of a building and the outer air when the building is heated in cold weather, and explains that the Neutral Zone is that point in the building that is equal in pressure to that of the air outside.

It gives an example showing that the difference between the weight of the air inside a large building when heated to 70 degrees and the outside temperature at zero would equal 28½ tons.

Formulas as to how this difference can be figured are also given.

Various illustrations accompany the paper showing the effect of different openings on the location of the neutral zone. Sketches of four typical buildings are given showing how the location of the neutral zone would be effected by their shape, and calculations are given to show that in certain high buildings that the velocity of the air movement in zero weather might be as high as 45 miles per hour.

DISCUSSION

Mr. Driscoll: The problem of locating the neutral zone seems to me to be almost impossible of solution from a practical standpoint and yet, paradoxical as it may seem, its possible location or locations should be theoretically determined if we are to secure the best results in heating or ventilating a tall building.

Its location would depend upon so many conditions that I believe it would be impossible to predetermine accurately its effect with relation to the amount of radiating surface or the ventilation of the building, and yet a study of the subject along the lines outlined in the paper would enable the engineer to avoid the fatal error of an insufficiency of either and at the same time he could scarcely avoid the error of an excess in some sections.

The author, in citing certain cases, states that: "These conditions are based on a condition of still air and tightly sealed

enclosing surfaces," and he further states that leaky or poor construction may have some effect on this neutral zone.

There are other conditions which to my mind affect the location of the neutral zone far more seriously than the conditions referred to and no mention has been made of them by the author.

In figures 12, 13, 14 and 15, the author indicates a single location in each type of building as the neutral zone and these locations are apparently correctly determined from the formulae given in the preliminary pages. They would only be correct, however, for theoretical conditions in which the buildings are constructed without floors, partitions, stairs, elevators, or interior construction and the effect of the installation of these parts would be to completely upset the results arrived at by the author's solution of the problem.

I do not think that there is any single point in a building which we could call the neutral zone, but there would probably be a number of points which would be practically neutral, and the points at which the engineer must fortify the building against an insufficiency of surface are at the points of inleakage which would naturally be below the neutral points.

Except in the few cases of tall buildings which are heated by indirect systems, which cases are so rare as to be counted as curiosities, no consideration need be given to the assumed outflow of air from the upper stories and the reduction of radiation surface on these floors for this cause would not be good engineering as the excess volume of air leaking in below finds easier means of exit than through the closed doors and windows on the upper floors. Stair and elevator hatchways leading to ventilators above corridor windows, etc., usually provide sufficient outlets to relieve the pressures within.

The greatest leakages that occur in the taller buildings take place on the first floor only, except in some rare instances where the arcade within extends through two or three floors. In such cases these floors are seriously affected by leakage and additional radiation must be provided for them.

Above these floors the conditions are practically neutral and this is particularly true where the floors are divided into numerous small rooms. In such cases the amount of radiation remains the same throughout for rooms of similar sizes, exposures and construction, the difference in the amount of leakage in or out not affecting the heating materially.

Where floors are undivided into rooms, however, the influence of the drafts in elevator hatchways and stair halls may be

felt to an appreciable extent and a slightly greater amount of radiation should be installed to provide for same. Such cases as this, however, are rare in the taller buildings.

An illustration of the conditions considered in this paper may be found in the McAlpin Hotel in New York.

The main lobby extends through two stories and has street entrances on the north and south. It also has direct connection to main stair halls and to all elevators and the terrific draft created in these hatchways exerts a tremendous "pull" on the lobby. The leakage through the main entrances north and south is reduced by the use of revolving doors. In some of the shops opening into the lobby, however, from the west it was impossible to install revolving doors. In the northwest corner of the building is located a drug store with three double doors leading to the street and two single doors opening on the hotel lobby. The radiation for this store was calculated in the usual manner and apparently ample allowances were made for the assumed extraordinary leakages which would take place on account of the constant opening of the street doors by patrons of the store. When the store was opened, however, it was found that not enough consideration was given to the additional leakage due to the tremendous "pull" in the lobby. The radiation in the store was doubled but only a partial cure was effected and the condition was finally overcome by placing revolving doors on the doors leading to the hotel lobby. I might add that the radiation was calculated by the designers of the store fixtures and was considerably less than that allowed by the engineer who designed the balance of the plant. It was determined, I believe, from the surface in a well heated store of similar dimensions operated by the same concern in another city but a consideration of the influence of the draft in the interior of the tall building was entirely neglected.

In the balance of the building, however, the conditions are practically neutral and the same amount of radiation suffices for the 4th floor as it does for the 24th floor and for all intervening floors, indicating that the leakage of air in or out of these floors is not sufficient to materially affect the heating. When we divide a building into rooms or offices and when we put a number of partitions between the outer walls and the stairways or elevator hatchways we overcome to a considerable extent the leakage that would take place through the windows and the more partitions we put in the less leakage we have. As a matter of fact I believe from my observation of the matter

where a building is divided up with a number of small private offices or rooms around the outside walls of the building considerably less radiation will be required to heat the building than if it were simply an open floor and the only reason for this condition is that we have reduced the leakage by reducing the possibility of circulation within the building and have established a practically neutral condition on each floor.

We get neutral zones at different points throughout the building and not at any single point and the locations of these zones are dependent upon whether the floors are with or without partitions or whether or not they are in the direct pathway of the currents which rush into the lower floors and up the elevator hatchways. The pressure which the author assumes we will get against the roof of the building is relieved, however, by reason of the fact that the air rushing up the elevator shafts secures free escape through the elevator machinery rooms which are generally open to the outside through ventilators or windows.

The idea that less radiation may installed in upper floors by reason of the outflow of air is to my mind erroneous for the reason that even if the heating of these floors was favorably affected by this outflow it would be more than offset by the additional heat losses on these floors due to the higher wind velocities at this elevation.

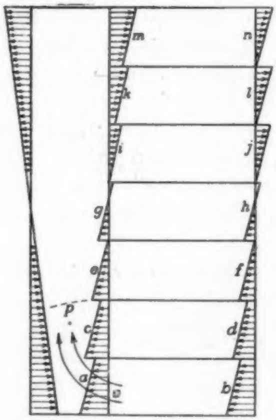
The paper is particularly interesting for the reason that it contains many thoughts that have never been the subject of discussion at the Society meetings within my recollection and will undoubtedly lead the way to considerable further investigation and discussion of a subject of extreme importance. The theories advanced by the author must be considered by the engineer because the heating of tall buildings is seriously affected by leakages and considerable thought must be given to the subject by engineers engaged in the heating and ventilating of tall buildings.

Mr. Lyle: I want to ask Mr. Blackmore if the building that started Mr. Ohmes in this investigation, was not the heating of some hotel on Fifth Avenue, that is heated with a blower system, and they have troubles of this kind, although they had the blowers every 3rd floor?

Mr. Blackmore: That particular building was not under investigation as a subject for this paper, but I hope some time to be able to get Mr. Ohmes to give us a description of the experiments he made in his investigation of that particular building.

He considers the information obtained in that investigation as confidential and not subject to publication at present.

I will make a sketch on the blackboard to illustrate some of the conditions Mr. Driscoll refers to, for while the location of the neutral zone is an elusive problem, I believe it is a problem that engineers must solve as nearly as it can be solved in any building they may have under construction.



We will assume that the sketch is a cross section of a building of seven stories, with an open stair or elevator hall from the ground floor to the roof. In such a building the neutral zone would be differently located on each floor due to the influence of the hallway.

If the conditions were the same on each floor the location of the neutral zone would be approximately as shown by the sketch. The ground floor zones would be about at a, b. The next floor at c, d and so on.

If there is a street door in the ground floor and an opening into the hallway at O, the condition mentioned by Mr. Driscoll would obtain, in which case the neutral zone might be taken entirely out of the ground floor room and equalize in the hall about at dotted line P, the exact location varying up or down with the difference in temperature between the inside and outside air and the size of the openings entering into and out of the ground floor room.

The opening of a door in cold weather from any of the floors into the hall would change the location of the zone as also would the opening of a window on any of the floors.

The paper was written with the idea of bringing to the notice of the members the necessity of considering the pressure differences in buildings when designing heating and ventilating plants.

Mr. Bushnell: Mr. President, this subject may seem to some rather academic in character but I am inclined to think that it may be found to have considerable to do with actual every-day heating conditions.

About five years ago we had a case in Chicago which illustrates the importance of this matter. We were heating a large office building nineteen stories in height and found that the heating requirements were very much greater than our estimate based on the theoretical heat losses from the exposed surface of the building. After puzzling over it for some time, we noticed that there was quite a strong draft in a pipe shaft which connected with the basement and ran clear to the roof. In this connection we noticed quite a pressure from the outside air at the first floor.

Some of our testing engineers were sent over to the building with an anemometer and the amount of warm air and heat units passing up the shaft was carefully estimated. As a result of that estimate we installed in the shaft some horizontal shutters or doors controlled by ropes passing over pulleys. In cold weather these shutters were kept closed with a resultant saving of from two to three thousand dollars per year in the cost of heating the building.

Mr. Murphy: I happened to have my offices for some time in a building that offers a very interesting check on Mr. Blackmore's neutral zone. The building is the Real Estate Trust Building in Philadelphia, and is similar to figure 12, except that it is 17 stories high. An attempt to heat this building with a blower system has never been satisfactory due to the fact that the neutral zone is in the neighborhood of the 9th floor.

The building is divided into very small offices, and there is a thermostat in each room, regulating volume dampers in the registers.

On the upper floors, when these dampers are opened by the thermostats the air rushes in with such tremendous velocity as to blow the papers off one's desk, while along the middle of the

building the heating is fairly satisfactory, on the lower floors it was difficult to heat the offices. They found these defects so serious that they are now putting in direct radiation, and trying to proportion it according to the location of the room; that is, using considerably more radiation at the bottom of the buildings than they are the top. If I remember correctly, the engineers told me that the lower floors of the building needed air at 39 degrees higher temperature in zero weather than the upper floors.

CCCLXXXVII.

ENGINEERING DATA FOR DESIGNING FURNACE HEATING SYSTEMS

ARTHUR C. WILLARD*

The designer of gravity furnace heating is confronted at the outset with a varied assortment of "data," largely empirical, more or less contradictory, and varying between wide and often indefinite limits. Even such of the data as has been derived from rational theoretical formulae applies only to isolated portions of the equipment, such as velocities in vertical flues as applied to "stacks," and not to the operation of the system as a complete unit.

The manufacturers of other heating equipment such as boilers, pumps, fans, blast coils, motors, etc., impose very definite limiting conditions upon the designer who uses their apparatus, and they have gone to great expense to secure such specific data from actual tests under service conditions.

Any successful method of testing furnaces for rating and performance must correspond with actual service requirements, and at the same time be capable of application by any progressive manufacturer, who is willing to put usable facts concerning his products in the hands of the designer. These tests must depend upon (1) the units in which the furnace is to be rated and (2) the data required in selecting the furnace and proportioning the system.

The furnace system is the simplest system that can be installed, in that the attendance required is a minimum not only during the heating season, but at the beginning or the end of the season, when the system has to be either made ready for or withdrawn from service. On account of its simplicity and the absence of water in the system there is no danger from freezing

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or explosion should the apparatus be left without fire, or fired and forgotten.

If properly designed the furnace system is very responsive to demands for an increase or decrease in the heat supplied since circulation starts or stops immediately the fire is forced or checked in accordance with the outside temperature requirements.

It is, of course, possible that if a furnace is improperly installed or operated the joints may open and permit flue gas to escape into the warm air space and vitiate the air supply.

The final consideration in making a choice of any heating system is always one of cost—cost to install—cost to operate—cost to maintain. Unfortunately the furnace system is supposed to be a very inexpensive system to install, much cheaper than steam or hot water and hence its selection is often decided upon when the funds available are actually insufficient to install any system. A furnace system selected for such a reason is bound to prove a disappointment to the owner. A satisfactory furnace system will cost to install, about 75 per cent. as much as a one pipe direct steam system and about 60 per cent. as much as a direct hot water system.

Operating cost will depend on the degree of ventilation required with the furnace system. If all the air for the latter is recirculated from within the house it will cost no more than a direct hot water system, which is capable of the greatest economy in operation of any method of heating.

The application of gravity steam and hot water systems to buildings of unusual shape or size is practically unlimited provided that considerable ventilation is not required. The field of the gravity furnace system is mostly confined to small buildings such as residences, stores, small schools, churches and halls which are ordinarily not above three or four stories high.

DATA REQUIRED FOR THE DESIGN

1. The total maximum B.t.u. transmitted to the air passing through the furnace per hour for standard initial and final temperatures, such as 32 deg. at entrance and 180 deg. at the furnace bonnet, when burning coal at a reasonable and practicable rate.

2. The efficiency of the furnace or ratio of the heat absorbed or transmitted to the air per hour above to the heat value of the coal burned per hour.

3. The allowable maximum combustion rate in pounds per square foot of grate using stove size anthracite for hard coal, and Pocahontas lump for soft coal furnaces. This will range from 5 to 8 pounds, depending on grate size, size and height of chimney, and size and length of smoke pipe connection. The determination of the flue gas temperature at the maximum rate would also prove of value in comparing furnaces.

4. Air temperatures at registers and data showing average loss in temperature between furnace bonnet and register for 1st, 2nd and 3rd floors.

5. Actual velocity in stacks to 1st, 2nd and 3rd floors when air in furnace bonnet is 180 deg. and the outside air is 32 deg., or less, with these stacks connected to leaders and registers of approved or average construction.

APPLICATION OF THE DATA

The application of the above data to the rating of a warm air furnace, and to the design of a gravity furnace system will be briefly outlined, and discussed in the following paragraphs:

I. *The furnace shall be rated* in B.t.u. transmitted to the air per hour, when the entering air temperature is 32 deg. and the air leaving the bonnet at the furnace is 180 deg. F. Below 32 deg. partial recirculation to be employed in order to secure an economy of operation comparable with direct steam and hot water systems. It will be seen that the suggested furnace rating is based on raising the temperature of each pound of air heated by approximately 150 deg. The square feet of grate surface and heating surface should also be given, as well as a dimensioned sketch of the fire pot and the fire pot should be of sufficient capacity to contain fuel for an eight hour charge with 20 per cent. left over to ignite the fresh fuel.

II. *The allowable maximum temperature* for rating and designing are to be 180 deg. at the bonnet, $(180 - t_d)$ at the registers, and 32 deg. at the cold air inlet. $t_d = 30$ deg. approximately = loss in temperature between bonnet and register, and $(180 - t_d) = 150$ deg. approximately. At present there is the widest variation in the temperatures used by different designers; those at the register, for example, ranging from 120 deg. to 160 deg. This results, not only in a variation of nearly 100 per cent. in the air volume required for a given room, but if recirculation is not practiced, a very great variation in the amount of heat is required from the furnace.

The loss in temperature between furnace bonnet and register is much greater than generally believed, and recent tests at the University of Illinois, on double wall stacks show that this loss may amount to from 20 deg. to 25 deg. in a 15-foot length depending on the velocity and temperature maintained. This loss of heat is almost always a direct demand on the furnace which must be very carefully allowed for in addition to the heat loss from the building, since the stack and leader loss is seldom of any value to the room being supplied by it.

III. *The heat loss from the building,* including leakage or infiltration, is to be computed in B.t.u. per hour and the total quantity of air, and air supply for each room, size of grate, etc., is to be based on this loss. This heat loss includes: (1) Heat transmission through walls and windows that may be figured in the usual way. (2) Loss due to infiltration through cracks around openings $= I \times L \times (t_1 - t_o)$ where $I =$ B.t.u. per lineal foot of crack at opening per 1 deg. This averages 1.2 B.t.u. per hour for good construction (1/32 inch crack) and 2.4 B.t.u. for poor construction (1/16 inch crack) with average wind velocities of 13 miles per hour.

$L =$ lineal feet around movable sash or doors.

t_1 and t_o internal and external air temperatures respectively.

NOTE:—Use the side of room having maximum length of crack, and neglect others.

IV. *Air required for each room* is to be based upon heat loss, and entering temperature of $(180 - t_a)$ at the registers, usually 150 deg. Where additional air for ventilation is demanded it must be introduced at a lower temperature than 150 deg., to prevent overheating.

V. *Register sizes* must depend on quantity of air passing through the grille, the allowable free area velocity, and the net free area. The allowable velocity for floor, base board, and low side wall registers is placed at 3 feet per second. The net free area depends on the grille design and varies from 25 per cent. to 75 per cent. of the register opening, averaging about 50 per cent.

VI. *Stacks* are the most difficult part of the system to standardize, due to the variety of shapes, possible locations, and special fittings such as heads and boots, which are used in connecting them to registers and leaders.

Since the quantity of air to be handled, and the height of stack are known the size can be determined *provided the velocity of flow is known.* It is at this point that we must turn to service tests of the complete system, if reliable data is to be available.

The theoretical velocity in a stack of given height with free inlet and discharge is easily determined, but is of no practical use when this stack is connected to a stack boot and leader at one end, and register box and grille at the other. The determination of stack velocities requires the most careful investigation. Commonly accepted values in use to-day are: $V_1 = 4.5$ ft. $V_2 = 6.5$ ft. and $V_3 = 8$ feet per second.

From the calculated areas leader diameters or stack proportions may be readily selected. Rectangular stacks with cross-sectional dimensions in a ratio greater than 3:1 are not effective over the entire area and hence should be avoided, or larger stacks used. Stacks are to be run in inside partitions, and covered with asbestos paper or made with double walls.

VII. *Leader sizes* should be based upon stack sizes, and made approximately 10 per cent. greater in area to allow for the loss due to higher air temperature in the leaders. All leaders should be graded uniformly 1" in 1'-0", and should have diameters increased ½" for each 5'-0" for lengths over 12'-0", or for elbows of 45 deg. or more. Leaders are to be made round in section and covered with three-ply asbestos air cell sheathing, approximately ¼" in thickness.

VIII. *Cold air and recirculating ducts* should be provided for all installations, and the cross sectional area made equal to the total stack area as already determined. A cold air room *without filters* (which cause loss of head, and soon prove unsanitary), is advisable. Its area should be four times that of the cold air duct. The return air register face should be kept out of the floor if possible, and its net free area made the same as the area of the return air duct.

IX. *Vent flues,* if necessary, are to be made of such size that they will be capable of removing enough air to supply the minimum ventilation requirements when the inside temperature is 40 deg. above the outside. Velocities in the same may be taken as 50 per cent. of the theoretical velocity based on height of flue, for the temperature range stated above. These flues are to be run in inside partitions and connected to an active roof ventilator. When the space around the chimney flue is made into an aspirating shaft the difference between the inside and outside temperatures may be taken as 100 deg. and 50 per cent. of the theoretical velocity used to determine the area. Toilet rooms and kitchens should always be vented, as well as important living rooms.

X. *Water pans or humidifiers* of the positive type only, with direct heating surface should be installed, and should have an evaporative capacity sufficient to maintain a relative humidity from 40 to 50 per cent. at 70 deg., when the air enters at 32 deg. and 25 per cent. saturated. Such pans or tanks must have an automatic feed under ball cock control in order to be effective. When the outside temperature approaches zero the interior humidity must be reduced to 40 per cent. or less to avoid condensation on cold surfaces.

XI. *Grate areas* are to be based on the total heat loss plus leader and stack transmission losses, using a maximum hourly combustion rate ranging from 5 to 8 lbs., of coal per square foot of grate, varying with size of grate. The range of efficiencies, usually 55 to 65 per cent., as found in rating tests, should be given and then the grate area can be easily determined.

XII. *Heating surface* must be so proportioned with respect to grate surface as to make it possible to heat all the air passing over the former to 180 deg., and at the same time have no part of the surface at a visible red heat. Since direct surface is far more valuable than indirect, and since a scrubbing or inpinging action of the air on one side and the hot gas on the other increases transmission, it is impossible to restrict this ratio within narrow limits. Ratios of to-day range from 12 to 1 to 40 to 1, and average transmission values vary from 1,000 to 3,000 B.t.u. per square foot of heating surface per hour. These transmission values are based on total heating surface, which includes direct and indirect taken together.

The engineering data required as indicated in the above discussion must be sought for and verified by careful tests made upon complete warm air furnace systems operating by gravity under service conditions.

ABSTRACT OF PAPER

The paper explains the need for more accurate engineering data for designing furnace heating and ventilating.

Explains how manufacturers of other appliances insist on definite restrictions about the placing of their appliances, but this seems to have been neglected by the furnace manufacturers.

The article also deals with the need for and urges the manufacturers to make experiments under service conditions to get some of the accurate data needed.

DISCUSSION

The President: The discussion on this paper will be opened by Mr. Chew.

Mr. Chew: Mr. President, I want to do something more than discuss this paper, I want to say something about the conditions in the Furnace Heating Industry.

The President asked two other members and myself to meet the National Furnace and Heating industry in Detroit, in June, and there I impressed the manufacturers with the fact that this Society since 1895 had been working in the interests of the Furnace Heating industry. I told them some 13 different furnace papers had been read before the Society, hence for 13 years out of the 20, this Society had discussed hot air furnace problems.

I want to say a word about the guide book published by the Fire Protection Association, in it they have put too many restrictions around the installation of hot air furnace pipes. They should be made to change them.

They have a very large influence. I have to take off my hat to the influence, because I cannot fight it, in every instance.

Professor Willard says in his paper that "leader sizes should be based upon stack sizes, and made approximately 10 per cent. greater in area to allow for the loss due to higher air temperature in the leaders." The wall stack now made double, to suit insurance companies, and to suit some manufacturers, will have a pipe with as much as 113 inches of area leading from the furnace to the stack it connects with that is only 3 x 10 inches, sometimes only 2½ x 10 inches to take its capacity.

For many years Professor Carpenter gave attention to the furnace heating industry, and seldom came to a meeting that he did not ask me—"What experiences have you had under your notice since I last saw you?" and a great deal that Carpenter has in his book on furnace heating, indirectly, is the result of conferences with me.

The furnace meeting I have referred to, where I represented the Society, Professor Willard read a paper to the effect that the Association should go on record in favor of refusing to put pipes in 4-inch studs wherever wall stacks were used.

In the early days, they used flues which ran to the basement and first floor and dining-rooms all in one chimney, and they ran from 8 inches square to as much as 9 x 12 inside, and they were better as warm air conduits than the smaller line flues,

double stack 3 x 10 or that are called for by the Fire Underwriters.

Professor Hoffman (written): We cannot too strongly emphasize the need for larger wall stacks in furnace systems. My views have been expressed fairly strongly in my hand-book.

"A possible improvement would be for the architect to anticipate the need for larger risers and provide suitable partition walls so that ample stack area could be put in. The ideal conditions will be reached when the architect actually provides air shafts of sufficient size to accommodate either round or a nearly square stack. When this time comes a great many of the furnace heating difficulties will have been solved."

There are moderate sized, second floor rooms, favorably located above the furnace and well protected, where a 4-inch studded wall stack will give entire satisfaction. The ordinary 10 x 12 or 12 x 15 room, moderately exposed, however, should have a 6-inch studded wall stack for good service.

Mr. Armagnac: There is another phase of this paper that is interesting, it says "Water pans or humidifiers of the positive type only, with direct heating surface should be installed, and they should have an evaporative capacity sufficient to maintain a relative humidity from 40 to 50 per cent., at 70 degrees, when the air enters at 32 degrees, and 25 per cent. saturated.

I would like to know if anybody in the room can tell us whether the warm air furnaces on the market are fitted with humidifying apparatus that will give that percentage of moisture?

Mr. Cooley: In that regard, I believe that some of the past equipment has not been put in the right place. The water appliance and humidifiers in furnaces are almost always put down near the floor where the cold air enters, and I believe if the same amount of surface, and the same amount of water was up where the hot air could reach it, and sufficient water would run to it, there would not be any doubt about getting the necessary humidity. I think there are one or two systems patented in which the humidifier was over the top of the furnace, and I do not think there is any doubt with that apparatus but what you could humidify sufficiently.

There is one other point in Mr. Willard's paper which should be taken up, which I think is very important, in regard to the hot air furnace, which is the location of the fresh air inlet, the direction of the compass in which the inlet should face. He says nothing whatsoever about it. If the inlet face to the

south, and the prevailing wind was from the north, it would undoubtedly result in an unsatisfactory operation; as the wind would be sufficient to increase the pressure on the north side of the house to such an extent as to prevent the entrance of air into the fresh air inlet. That does not seem to be brought up here.

I think an ideal furnace would be one in which it had a fresh air inlet facing all four points of the compass, with a flap damper in each one of them, so that no matter where the wind blows, it would always blow into the fresh air chamber.

CCCLXXXVIII.

THE ESTABLISHMENT OF STANDARD METHODS OF PROPORTIONING DIRECT RADIATION AND STANDARD SIZES OF STEAM AND RETURN MAINS*

BY JAMES A. DONNELLY

HEAT LOSSES FROM BUILDINGS

The amount of heat lost from the several rooms of a building is computed from the sum of the heat losses transmitted through the windows, walls, etc., and the heat carried away by the air used for ventilation or entering by leakage around windows, through walls, etc.

All heat losses are tabulated in B.t.u.'s per square foot per hour for what is considered as the standard condition of Zero outside and 70 degrees inside. The proportionate heat loss for other than standard conditions will afterward be tabulated.

Single windows or skylight	70
Double windows or skylight	40
Tar or gravel roof	20
Mill construction, T. and G.	15
Concrete with cinder fill	30
Slate sheathed	26
Tin sheathed	22
Shingle sheathed	20

The following table of losses through brick and concrete walls are based on the tests of L. A. Harding, which tend to show that the rate of transmission is inversely proportional to the thickness.

	Brick Wall	Brick Plastered	Furred and Plastered	Concrete Wall	B. t. u.
8½ in.	41	35	29	4 in.	75
13 in.	27	24	21	6 in.	50
17½ in.	20	18	16	8 in.	37
22 in.	16	15	14	12 in.	25
26½ in.	13	12	11	16 in.	18

* This paper is part of a report of the joint educational committee of the National District Heating Association and this Society read at their annual convention in Chicago, June, 1915.

For ceilings over 12 feet add 2 per cent. for each extra foot in height.

For frame buildings, lathed and plastered on the inside with the outside as below:

Ordinary clapboards	33
Ordinary clapboards paper lined	24
Ordinary clapboards sheathed	21
Ordinary clapboards sheathed and paper lined	18
Ordinary clapboards sheathed and back plastered	15

The temperature of unheated spaces, such as attics, entrance halls, cellars, etc., is usually considered as about midway between the outside and inside temperatures, depending somewhat on the location and construction. These heat losses are, therefore, given in B.t.u.'s per square foot per degree.

Partitions between heated and unheated rooms:	
Lath and plaster one side	.6
Lath and plaster two sides	.3
Floors between heated and unheated rooms:	
Single wood flooring	.3
Single wood flooring, plaster below	.2
Ceilings between heated and unheated rooms:	
Lath and plaster	.6
Lath and plaster, floor above	.45

The basis for estimating the air required for ventilation is the number of cubic feet that should be introduced at the room temperature, in order to maintain a certain standard of purity of the air, or in other words, good ventilation. In the same way the basis of computation for the leakage of air into a room is the number of cubic feet of space that this air will occupy when heated to 70 degrees. One cubic foot of air at zero weighs .086 pounds, at 70 degrees it weighs .075 pounds. Thus if the leakage of air into a room at zero is .87 cubic feet, this amount of air when heated to 70 degrees will expand to 1 cubic foot. The weight of .87 of 1 cubic foot of air at zero is .075 pounds; this multiplied by the specific heat of air at constant pressure, .2375, equals .0178 B.t.u. to raise this amount of air 1 degree. This multiplied by 70 gives a total of 1.246 B.t.u. to raise the air necessary to supply one cubic foot of ventilation per hour from zero to 70 degrees.

Therefore, the cubic contents of a room must be multiplied by 1.25 for each change of air per hour by leakage, or for ventilation.

The amount of air leakage is difficult to estimate accurately and can only be approximated. The standard allowances are usually:

Factories, large lofts, etc.	¾ to 1½	changes
Living rooms with doors usually open	1½	changes
Living rooms	1	change
Living rooms with open fire-places	2	changes
Bed-rooms, office rooms, etc.	1	change
Entrance halls	3	changes

It has become customary to add 10 per cent. to the total heat losses of a room for north or west exposures; 15 per cent. for those heated in the day-time only and 25 per cent. for those heated only at long intervals.

A close approximation of the total heat losses from a building cannot be made merely by the application of the foregoing or any other set of theoretical values. A considerable amount of judgment and experience must also be available. The quality of building construction will vary within wide limits, even when erected under the same or very similar specifications. The character of buildings of the same class is also materially changing. In some cases it is deteriorating, in other cases it is improving. Speculative suburban residences have much higher heat losses than a residence in which the conditions are apparently similar, but which is erected for investment or to order, under competent and efficient supervision.

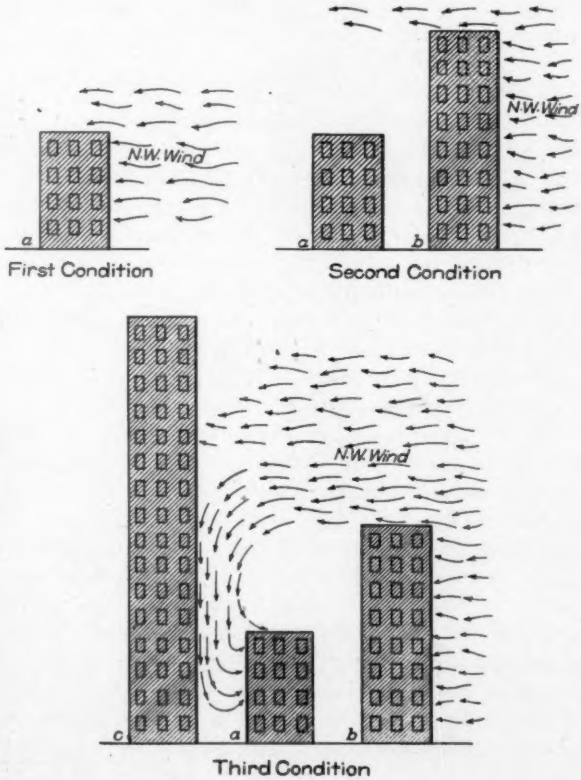
In non-fireproof factories with open stairways, it was formerly not necessary to figure the roof exposure, as the heated air rising from the lower floors was sufficient to overcome it, and the top floor did not usually require any more radiation than the intermediate floors. However, the development of fireproof, slow burning or mill construction factories with tightly enclosed stairways and elevator shafts has, by cutting off this upward current of air, rendered it necessary to make close studies of the losses by roof exposures.

The same condition exists in residences. Where all room doors are open, the air leakage will be found to be inward through the crevices of the lower floors, up the stairways, and outward through the crevices of the upper floors, attic, eaves, etc. In high buildings this has led to the necessity of more radiation on the lower floors, the normal amount at the neutral zone, near the middle of the building, and a lessened amount on the upper floors. However as it is desirable from a hygienic as well as a fire pre-

vention standpoint, to close off the air leakage from floor to floor, there has been a gradual improvement in this respect, and a very decided change for the better in many buildings.

In factory buildings exposed on all sides and without tight dividing partitions, there will naturally be a current of air through the building corresponding to the wind direction outside. It is, therefore, good practice in such buildings to reduce the theoretical amount of radiation for the south and east sides and increase it on the north and west sides, or the sides of the prevailing cold winds.

The position of a building in its relation to other buildings often affects it very seriously and the further erection of buildings sometimes reverses the former conditions. To illustrate, a is the elevation of a building four stories high, exposed to the right which is northwest. Some time after it is built, the building b



is erected, eight stories high, which completely protects it from the direct action of the northwest winds. Later the building c is erected, sixteen stories high. The northwest wind impinging against the top of this building rebounds against the building a, thus almost completely reversing the original condition.

Tight construction and good weather stripping are of great importance, and it should be remembered that the warm side of a building should be weather-stripped in order to prevent the warm air from going out as well as the cold side to prevent the cold air from coming in. It is better to have a building tightly constructed and well weather-stripped and slightly open the windows for ventilation, if no other means is available, than to have a leaky building and ample radiation. For the former building will hold its temperature very much longer at night, when heat has been shut off, and will be very much easier maintained at any desired temperature during non-occupied hours. The cooling curve of a building is one of the best tests of quality of construction.

TRANSMISSION FROM RADIATING SURFACES

The heat transmitted by various direct radiating surfaces has been the subject of many tests and may be very accurately determined, as the possible variables are comparatively few. All tests, should, of course, be conducted as nearly as possible under the conditions and in the same manner in which the devices are to be used. It is well to have a thermometer above and one below the radiator for recording the temperature of the air flowing to and from it; as these may sometimes be found to vary while the temperature of the room may be constant.

The pressure usually carried on low pressure steam plants has been considerably reduced by the wide introduction of vacuum return line and vapor systems, thus perhaps slightly reducing the heat transmission. The completeness of air and water removal has, however, been materially improved, thus increasing the heat transmission. It is now customary to obtain a complete circulation with a pressure of one-quarter to one-half pound at the source of supply. This pressure, and the resultant temperature of the steam, are frequently reduced in the course of distribution so that the radiators are often operating at a temperature of not over 210 degrees. This temperature of the steam is therefore taken as the standard condition, and used in compiling the following table of the transmission of heat in British thermal units per square foot per hour, under standard conditions of room 70

degrees, radiation 210 degrees. The proportionate transmission for other than standard conditions will afterward be tabulated.

Wrought pipe coils 1 in. to 2 in. diameter....300.
Cast iron wall radiators, on side............290.
Cast iron wall radiators, on end.............280.

CAST IRON DIRECT RADIATORS

	45 in.	38 in.	32 in.	26 in.	22 in.
1 column	262	270	275	279	285
2 column	241	250	257	263	270
3 column	222	231	240	248	255
4 column	210	218	225	233	240

Direct radiators, provided with a box base, taking a fresh air connection from outside, and usually called direct-indirect, have frequently been used, but are not now considered very good practice. It is very difficult to make the box base sufficiently tight so that the cold air will not leak out in objectionable quantities. When they were placed on both the warm and the cold sides of a building, the air would enter on the cold or windward side, but would frequently flow out on the warm or leeward side.

The rate of transmission of heat from a direct radiator does not vary directly as the difference in temperature between the radiator and the room. The co-efficient per degree difference is found to vary approximately 2 per cent. for each ten degrees variation from standard conditions, as developed by C. A. Fuller.

The comparative transmission at various temperatures is shown in the following tables:

Steam Temp.	Room Temp.	Diff. Temp.	Standard co-eff.	B.t.u. per Sq. Ft. per Hr.	Proportionate Transmission
260	70	190x	1.785 plus 10%	373	1.49
250	70	180x	1.785 plus 8%	247	1.39
240	70	170x	1.785 plus 6%	322	1.29
230	70	160x	1.785 plus 4%	297	1.19
220	70	150x	1.785 plus 2%	273	1.09
210	70	140x	1.785 *standard*	250 *condition*	1.00
200	70	130x	1.785 minus 2%	227	.91
190	70	120x	1.785 minus 4%	206	.82
180	70	110x	1.785 minus 6%	185	.74
170	70	100x	1.785 minus 8%	164	.66
160	70	90x	1.785 minus 10%	145	.58

In order to find the temperature of a radiator for any desired transmission, the above process may be reversed by the application of the following formula: Temperature of radiator equals $\sqrt{280T \text{ plus } 32{,}400} - 110$. In which T equals transmission in B.t.u. per sq. ft. per hour.

Example: If the transmission of a radiator is 250 B.t.u. under standard conditions of room 70 degrees, radiator 210 degrees, what will be the temperature of the radiator for a transmission of 125 B.t.u. per sq. ft. per hour?

Solution: 280 × 125 equals 35,000, which added to 32,400 equals 67,400. Extracting the square root, gives 259.6, subtracting 110 leaves, 149.6 which is the required temperature of the radiator.

The following table gives the proportionate heat losses from buildings, with varying outside minimum temperatures, and with a room temperature of seventy degrees. Also the proportionate amount of radiation to heat the room to seventy degrees with steam at the radiator at 210 degrees.

Outside Minimum Temperature	Inside or Room Temperature	Radiation Required	Proportionate Heat Loss Proportionate
35	70	.50	.50
30	70	.57	.57
25	70	.64	.64
20	70	.71	.71
15	70	.79	.79
10	70	.86	.86
5	70	.93	.93
Zero *standard*	70	1.00 *conditions*	1.00
− 5	70	1.07	1.07
−10	70	1.14	1.14
−15	70	1.21	1.21
−20	70	1.29	

The proportionate surface required to heat a room to seventy degrees, under the above standard conditions for the radiator, is the same as the proportionate heat loss for any outside minimum temperature.

The following table gives the proportionate heat losses from buildings, the proportionate transmission from direct radiators, and the proportionate radiation required (with steam at 210 de-

grees) for various room temperatures, when the outside temperature is zero:

Proportionate Heat Losses from Buildings.		*Proportionate Trans. of Direct Radiators.*		*Proportionate Surface Required for Heating*	
Room Temp.	Proportionate Loss in B.t.u.	Diff. in Temp. bet. Radiator and Room	Proportionate Trans. in B.t.u.	Temp. Room	Proportionate Surface Req. Sq. Ft.
35	.50	175	1.34	35	.37
40	.57	170	1.29	40	.44
45	.64	165	1.24	45	.52
50	.71	160	1.19	50	.60
55	.79	155	1.14	55	.69
60	.86	150	1.09	60	.78
65	.93	145	1.05	65	.89
70	1.00 *standard*	140	1.00 *conditions*	70	1.00
75	1.07	135	.95	75	1.12
80	1.14	130	.91	80	1.26
85	1.21	125	.87	85	1.40
90	1.29	120	.82	90	1.56
95	1.36	115	.78	95	1.74
100	1.43	110	.74	100	1.93
105	1.50	105	.70	105	2.15
110	1.57	100	.66	110	2.39
115	1.64	95	.62	115	2.66
120	1.71	90	.58	120	2.95

Assuming that the rate of heat loss from a building varies directly with the difference between the outside temperature and the building temperature, and considering the heat loss for zero outside, 70 degrees inside, as the standard, or 100 per cent.; the second column shows the proportionate loss of heat from a building when the outside temperature is zero, and the inside temperature is as given in the first column.

Assuming that the rate of transmission from a direct radiator to the air of a building is in proportion to their difference in temperature, with a variation in the rate of transmission of 2 per cent., greater or less, for each 10 degrees increase or decrease in their temperature difference, and considering 140 degrees difference in temperature (steam 210 degrees, building 70 degrees) as standard, or 100 per cent. transmission, the second column shows the proportionate transmission when the difference in temperature is as given in the first column.

Assuming that under standard conditions of outside temperature zero, building temperature 70 degrees, and radiator temperature 210 degrees (or 140 degrees difference between the radiator and room) the amount of radiation necessary is 100 per cent., the proportionate amounts of radiation given in the second column, are those necessary to heat a building to the temperatures given in the first column, when the outside temperature is zero.

NOTE—The amount of surface required for heating is always obtained by dividing the heat loss from the building by the amount of heat transmitted per square foot of radiation. Therefore, as may be seen from the above tables, the proportionate amount of surface required for heating is obtained by dividing the proportionate heat loss from the building by the proportionate transmission of the radiator, in each case.

The preceding table may be used to find the proportionate amount of radiation necessary to heat a room to any desired inside temperature, other than 70 degrees, when the outside minimum temperature is other than zero. Find the difference between the outside temperature and the room temperature in the first column; divide the proportionate heat loss opposite this amount, in the second column, by the proportionate transmission opposite the difference in temperature between the radiator and the room, as given in the fourth column, and the result will be the proportionate amount of radiation required.

Example: What is the proportionate amount of radiation required to heat a room to 90 degrees, with a temperature of 20 degrees below zero outside?

Solution: The difference in temperature between 20 degrees below outside, and 90 degrees inside, is 110 degrees. Opposite 110, the proportionate heat loss or 1.57 is found in the second column. The difference in temperature between the radiator and the room (steam 210 degrees, room 90 degrees) is 120 degrees. Opposite this the proportionate transmission, .82 is found in the fourth column. Divide 1.57 by .82 and the quotient, 1.91 is the proportionate amount of radiation required.

COMPARISON OF OPERATING CONDITIONS

Gravity hot water systems using an open tank are usually figured for a maximum temperature of about 180 degrees at the boiler, with a drop of 20 degrees through the system, thus having 160 degrees in the return at the boiler, or an average temperature at the radiators of 170 degrees. Pressure water systems, using ordinary safety valves or mercury seals, as well as forced circulation systems (those using circulating pumps), are often figured as high as 220 degrees to 230 degrees, thus frequently maintaining an average for the radiators of 210 degrees in zero weather.

Some attempts have been made to operate an entire heating plant under a steam pressure below that of the atmosphere in mild weather, varying the temperature of the radiator by the vacuum carried, in proper relation to the outside temperature; so that, with the entire radiator heated, its temperature would be at the proper point for maintaining the room temperature at 70 degrees. In this case the various radiator temperatures would be the same as for a forced circulation hot water system, provided the radiator temperature for zero weather was the same (210 degrees) for each system.

The following table gives the required temperature of the radiator for various outside temperatures for a forced water circulation or a vacuum steam system; the pressure of steam or the degree of vacuum which must be carried to maintain this temperature with the steam system and the B.t.u. transmission by a standard cast iron radiator under this temperature. Also the required temperature of the radiator and the transmission for a gravity circulation water system. In the last column, the proportional amount of the radiator which is to be heated for various outside weather conditions is given for a fractional air-return vacuum or vapor system.

	Forced Water or Vacuum			*Gravity Water*		
Outside Temp.	Temp. of Rad.	Pressure or Vacuum of	B.t.u. per sq. ft.	Temp. of Rad.	B.t.u. per sq. ft.	Percentage Rad. Frac. System
–20	240	10.25 lbs.	322	192	211	1.29
–15	233	7.25 lbs.	304	187	199	1.21
–10	225	4.25 lbs.	286	181	187	1.14
– 5	218	2.0 lbs.	268	176	176	1.07
Zero	210	1.0 in. vac.	250	170	164	1.00
5	202	5.5 in. vac.	232	162	149	.93
10	194	9.0 in. vac.	214	157	138	.86
15	186	12.5 in. vac.	196	150	126	.79
20	177	15.5 in. vac.	179	144	115	.71
25	168	18.0 in. vac.	161	137	103	.64
30	159	20.5 in. vac.	143	131	92	.57
35	150	22.5 in. vac.	125	124	80	.50
40	140	24.0 in. vac.	107	118	69	.43
45	130	25.0 in. vac.	89	110	57	.36
50	119	26.5 in. vac.	71	103	46	.29
55	108	27.5 in. vac.	54	95	34	.21
60	96	28.3 in. vac.	36	87	23	.14
65	83	28.8 in. vac.	18	79	11	.07

It will be noted that the degree of vacuum which it is necessary to carry in order to properly heat a building, increases very rapidly for a comparatively slight rise in the outside temperature. This makes it practically impossible to carry a sufficiently low pressure and temperature to prevent overheating by this method alone. It is very difficult to maintain vacuums of even fifteen to twenty inches, and almost impossible to maintain one higher than this. It has, therefore, come to be considered much better practice to control the heating effect of a radiator by heating a portion of it by means of a fractional system, rather than attempt to

heat all of the radiator but reduce its temperature in relation to the outside temperature by carrying a varying vacuum within it.

It will be seen from an examination of the column of radiator temperatures for a gravity water circulation, that where a system is designed for 170 degrees water at the radiator in zero weather, that if the heater is large enough to maintain the water at 192 degrees, the radiation would then be sufficient to keep the building at 70 degrees when the outside temperature was 20 degrees below zero.

In the same way it may be noted that where a fractional vapor or vacuum system is designed for substantially atmospheric pressure in zero weather, if the boiler is large enough and the system is of such a type that it may be run at 10 pounds steam pressure, the radiation, being 29 per cent. more efficient at this pressure, will suffice to heat the building to 70 degrees when the outside temperature is 20 degrees below zero.

In a moderate climate where the minimum temperature seldom drops below 10 degrees, it is quite safe to design a steam system for atmospheric pressure at about 16 degrees above zero outside temperature. This would make about ten pounds steam pressure necessary if the minimum of zero outside was ever reached. The standard hot water temperature may be shifted in the same manner. The result of these changes is shown in the following table:

	Forced Water or Vacuum Steam			*Gravity Water*	
Outside Temp.	Temp. of Rad.	Pressure or Vacuum of	B.t.u. per sq. ft.	Temp. of Rad.	B.t.u. per sq. ft.
Zero	240	10.25 lbs.	322	192	211
5	231	6.5 lbs.	299	185	196
10	221	3.0 lbs.	276	178	181
15	211	1.0 in. vac.	253	171	166
20	201	5.5 in. vac.	230	163	151
25	191	10.0 in. vac.	207	155	136
30	180	14.0 in. vac.	185	147	121
35	168	18.0 in. vac.	161	139	105.5
40	156	21.0 in. vac.	138	130	90
45	144	23.0 in. vac.	115	121	75
50	131	24.5 in. vac.	92	112	60
55	117	26.0 in. vac.	69	102	45
60	103	28.0 in. vac.	46	92	30
65	87	28.5 in. vac.	23	81	15

The steam boiler or hot water heater should not, however, be reduced in size when the amount of radiation is reduced in this manner. The size of the boiler should be proportioned from the B.t.u. losses of the building from the minimum outside temperature to 70 degrees inside.

The following table gives the proportionate radiation required to heat a building from zero outside to 70 degrees inside, with varying maximum temperatures of the radiators. Also the proportionate size of the boiler or heater for the corresponding maximum temperatures of the radiators:

Temp. of Radiator	Transmission	Proportionate Surface Required	Proportionate Boiler Load Steam	Prop. Load Hot Water
240	322	.78	1.29	....
230	297	.84	1.19	....
220	273	.92	1.09	....
210	250	1.00	1.00	1.52
200	227	1.10	....	1.38
190	206	1.21	....	1.26
180	185	1.35	....	1.13
170	164	1.52	....	1.00
160	145	1.72	....	.88
150	126	1.99	....	.77

STANDARD SIZES OF STEAM MAINS

The size of steam mains depends upon so many factors, that a complete discussion of the subject would require the presentation of preliminary matter covering almost all parts of a steam heating system. See papers by the author, published in the Society Transactions, Vol. 12, 1906, and Vol. 13, 1907.

However, when the steam is to be distributed at a comparatively low pressure to a number of buildings or through the basement of a building of large size, the amount of radiation that may be supplied for any drop not given in the table may be found as follows:

As the drop in pressure varies as the square of the velocity, one-half the velocity, or one-half the radiation to be supplied, equals one-quarter the drop; twice the velocity, or twice the radiation to be supplied, equals four times the drop, etc.

This is illustrated by the following table of the capacities of a one-inch pipe for various drops in pressure:

Drop	Square Root of	Equals	Multiply by	Equals Capacity
1/8 oz.	.125	.353	40	14.1
1/4 oz.	.25	.5	40	20.
1/2 oz.	.5	.7	40	28.
3/4 oz.	.75	.866	40	34.8
1 oz.	1.	1.	40	40.
1 1/4 oz.	1.25	1.118	40	44.7
1 1/2 oz.	1.5	1.225	40	49.
1 3/4 oz.	1.75	1.322	40	52.88
2 oz.	2.	1.414	40	56.56

For lengths of run other than 100 feet, multiply the drop in pressure by the ratio of the given run to 100.

Where the steam is to be distributed at high pressure and used through reducing valves at the buildings, the table of the Unwin formula as it has been divided for convenient figuring by W. L. Durand may be used. Column 1 gives the factor for various drops in pressure from one ounce to three pounds per 100 foot run. Column 2, the factor for the pipe size from 1 inch to 16 inches. Column 3, the factor for various densities of steam from atmosphere to 200 pounds gauge pressure. Therefore, if

TABLE OF FACTORS FOR THE UNWIN FORMULA. NOMINAL SIZES OF PIPE

Column One		*Column Two*		*Column Three*	
Drop in Press.	$87\sqrt{\frac{P}{100}}$	Dia. Pipe	$\sqrt{\frac{d^5}{1+\frac{3.6}{d}}}$	Steam Press. lb. per sq. in.	$\sqrt{D}$
1. oz.	2.17	1 in.	.466	0	.193
.1 lb.	2.75	1 1/4 in.	.887	2	.207
.2 lbs.	3.98	1 1/2 in.	1.49	5	.223
.3 lbs.	4.76	2 in.	3.38	10	.248
.4 lbs.	5.50	2 1/2 in.	6.32	15	.270
.5 lbs.	6.15	3 in.	10.5	20	.290
.6 lbs.	6.73	3 1/2 in.	16.1	30	.326
.7 lbs.	7.27	4 in..	23.	40	.358
.8 lbs.	7.78	4 1/2 in.	32.	50	.388
.9 lbs.	8.26	5 in.	43.	60	.415
1. lbs.	8.70	6 in.	70.	75	.452
1.25 lbs.	9.74	7 in.	105.	100	.508
1.5 lbs.	10.6	8 in.	150.	125	.557
1.75 lbs.	11.5	9 in.	205.	150	.603
2. lbs.	12.3	10 in.	271.	175	.645
2.5 lbs.	13.7	12 in.	437.	200	.684
3. lbs.	15.	14 in.	654.	. . .	. . . .
		16 in.	925.	. . .	. . . .

the proper factor is selected from each column, the product of the three will give the pounds of steam that will be delivered per minute. This amount multiplied by 200 will give the square feet of direct radiation that may be supplied by the steam.

Example: What is the weight of steam that will be delivered per minute through a 4-inch pipe with steam at 30 lbs. and a drop in pressure of 5 lbs. for a run of 500 feet?

Solution: The drop as given is equal to 1 lb. per 100 feet. From Column 1, the factor opposite 1 lb. drop 8.7 is obtained. From Column 2, opposite 4-inch pipe, 23. From Column 3, opposite 30 lbs. pressure, .326. The product of these three is 66.65 lbs. This multiplied by 60 equals 3999 lbs. per hour. Dividing by .3 lbs. per sq. ft. (which is a safe factor per square foot of direct radiation, including allowance for the condensing capacity of the distributing mains within the buildings) this gives 13,330 sq. ft. of radiation that may be supplied.

It is well to have a table of the comparative carrying capacity of pipes as figured from the Unwin formula, so that after one size is figured for a certain condition, the capacity of all other sizes may be very quickly obtained by it. The following table gives this information where the nominal sizes are used as a basis

TABLE OF THE COMPARATIVE CARRYING CAPACITY OF PIPES

UNWIN FORMULA

Nominal Diam	C.C.C.	Standard Diam.	C.C.C.	Ex. Heavy Diam.	C.C.C.	Dbl. Ex. Hy. Diam	C.C.C.
1	1.	1.048	1.15	.951	.865	.587	.212
1¼	1.9	1.380	2.53	1.272	2.	.885	.702
1½	3.21	1.611	3.94	1.494	3.17	1.008	1.275
2	7.25	2.067	8.07	1.933	6.98	1.491	3.15
2½	13.6	2.468	13.1	2.315	10.9	1.755	5.01
3	22.5	3.067	25.4	2.892	20.3	2.284	10.5
3½	34.5	3.548	35.7	3.358	30.8	2.716	17.1
4	49.2	4.026	50.7	3.818	43.9	3.136	25.5
4½	68.7	4.508	69.	4.280	59.9	3.564	36.4
5	91.4	5.045	93.6	4.813	82.4	4.063	52.
6	150.	6.065	154.	5.751	133.	4.875	85.4
7	226.	7.023	228.	6.625	195.	5.875	141.
8	322.	7.982	321.	7.625	291.	6.875	215.
9	440.	8.937	433.	8.625	393.	....	......
10	582.	10.019	584.	9.750	544.	....	......
12	938.	12.	938.	11.750	888.	....	......
14	1400.	14.	1400.	.....	......	....	......
16	1980.	16.	1980.	.....	......	....	......

for the calculation, as well as the comparative carrying capacity of standard, extra heavy and double extra heavy pipe.

Note—It will be seen from an examination of the above table that the difference in the capacity of pipes as figured on the nominal and actual diameters, is very little, except in the sizes of two-inch and below; where they are somewhat higher when figured on the actual diameters. The larger sizes are usually cut off with a knife cutter, which does not reduce the internal diameter, but it is customary to cut off the sizes below two-inch with a wheel cutter, which always reduces the diameter. Although all pipes that are cut with a wheel cutter should be afterward reamed so that the original diameter is restored, this is very often neglected entirely or else very insufficiently done. It is, therefore, considered better practice to use the table of nominal sizes in calculating the carrying capacity of wrought pipes.

STANDARD SIZES OF RADIATOR CONNECTIONS AND VALVES

The impartial and disinterested observation of radiator manufacturers upon the comparative working of radiators with different sizes of connections, has led to the very wide adoption of standard sizes of radiator connections for one and two-pipe gravity systems. They are given in the following table, as well as sizes for vapor and vacuum systems that are believed to represent the best of present practice.

	Supply				*Return*		
	One Pipe Steam	Two Pipe Steam	Vapor and Vacuum Steam Con.	Vapor and Vacuum Valve	Two Pipe Return	Vapor Return	Vacuum Return
½ in.	...	...	...	20	...	40	80
¾ in.	...	20	20	40	40	100	200
1 in.	24	40	40	80	80	200	400
1¼ in.	60	80	80	160	160	400	750
1½ in.	100	160	160	320	320	...	...
2 in.	200	320	320	...	...	...	...

The table given above shows the maximum amount of radiation that can be supplied or drained by the given sizes of pipe, for various systems of steam circulation. The run-outs and the valves are of the same size, with the exception of the steam pipe and valve for vapor and vacuum systems. In these it is customary to make the inlet valves smaller than the pipe connection;

the reduction in size from the standard sizes of steam piping to the smaller size of the valve is to be made as close to the radiator as possible.

STANDARD SIZES OF RETURN MAINS

Return mains may be divided into two classes; wet returns, or those that are below the water line, and dry returns, or those that are above the water line. Wet returns carry only the water of condensation; dry returns carry the condensation and also enough steam to supply the radiation from them.

Good engineering practice has determined that the velocity of flow through both kinds of returns, should not cause a higher pressure drop by friction than that recommended for standard sizes of steam mains; namely, one ounce to the one hundred feet in straight pipe.

The comparative flow of water and steam with the same drop in pressure from friction is proportionate to the square root of the density. As water at 212 degrees is about 1600 times as heavy as steam at atmospheric pressure, a wet return pipe will drain about forty times as much radiation as the same pipe will supply, with the same pressure drop from friction.

In figuring the proper rating for pipes as dry returns, it is necessary to assume how much surface they will have in order to know how much steam they will be required to carry in order to supply the heat radiated by them.

If 1,640 cubic feet of steam are fed into a radiator and 1,600 cubic feet are condensed and passed into the return, as well as 40 cubic feet of uncondensed steam, the size of the pipe required to carry either the water or the steam will be the same; since the steam is 40 times the volume of the water and flows at 40 times the velocity with the same pressure loss from friction.

From this it will be seen that the size of the pipe required to pass $2\frac{1}{2}$ per cent. of the amount of steam condensed by the radiator, is equal to that required by the water of condensation from the radiator (neglecting the advantage due to difference in elevation as regards the condensation) and the following formula is very readily deduced:

$$r \text{ equals } R \frac{40}{1 + 4N} \text{ in which,}$$

r equals the rating of a pipe as a wet or dry return.
R equals the steam main rating of the same size pipe.
N equals percentage of surface of radiation in exposed return surface.

If it is assumed that the radiating effect of the return piping is usually about 10 per cent. that of the radiation, then the return radiator connections and the return risers may be laid out from the 10 per cent. column and the return mains farthest from the source of supply from the 7½ per cent. column. The assumed percentage may gradually be reduced until finally when the return drops below the water line, it may be sized from the wet return column. If the returns are to be covered, due allowance may be made, and the sizes reduced accordingly.

ABSTRACT OF PAPER

The author explains how heat is lost through a building and gives tables showing losses through different materials.

Gives data on comparative values of radiators of different heights and widths. Explains how the coefficient of transmission varies through differences of temperature.

Deals with the size of main, flow and return pipes.

Gives tables setting forth capacities of different size pipes, based on the Unwin formula.

Also deals with the size of radiator valves and connections.

DISCUSSION

Thos. Tait: The paper by Mr. J. A. Donnelly on "The Establishment of Standard Methods of Proportioning Direct Radiation and Standard Sizes of Steam and Return Mains" is a very able article, but, judging from the experience of the writer, covering a period of over 25 years in various portions of the country, the intent of the paper, namely, establishing a standard for figuring radiation and pipe sizes, will never be realized.

Even though the Society took upon itself the responsibility or burden of establishing "standards" for the above items, it is doubtful if they would be used by the majority of those having to do with this class of building equipment. Many of you know the tendency of one man in the heating profession to decry the work of another, and often for no other reason than that the first man did not get the job. This is a deplorable fact, but true, nevertheless.

The majority of men engaged in this line of work consider that their own "standards" are plenty good enough for them, that they have seen the results obtained by their use, and will lead themselves to believe that any change will not be bene-

ficial. It will be a difficult matter to change the opinion of this class of men.

Mr. Donnelly has presented some very interesting data on pipe sizes, the first of which came to the attention of the writer in 1906, "Sizes of Return Pipes." The papers demonstrate the fact that Mr. Donnelly has given these items considerable time and attention.

Regarding the table of radiator connections, the writer is of the opinion that the sizes for one pipe steam are correct, but for two pipe steam, or "vapor," the sizes are larger than necessary.

The writer calls to mind a two pipe gravity steam job, using pressure reduced from 25 to 30 lbs. gauge to 1½ lbs. gauge, with returns running by gravity to an automatic receiver and then pumped back into boiler. Owing to limited wall space the coils had to be massed in large units, and one of the coils, having 456 sq. ft. of surface, was supplied through a 2-inch angle valve and the return end was fitted with a ¾-inch swing check valve. From the bottom of the manifold to the underground return in trench was 9 inches, yet the water never stood in the bottom of the coil or backed up from the main return. There were 8 coils under similar conditions but somewhat smaller.

The writer personally knows of a great many radiators of 100 sq. ft. capacity, installed in conjunction with one of the so-called "vapor" systems, not using a pump for handling the returns, where the supply is a ¾-inch valve restricted to an actual steam opening of ⅜ inch, with the return end fitted with a ½-inch valve, full opening, with a gage pressure of ½ lb. or 8 oz., giving the very best results.

The writer was called upon recently to estimate a two-pipe gravity steam job, for a school house, where the pressure was supposed to be 2 lb. gauge, where the specifications called for a 63 sq. ft. radiator to be supplied through a 1½-inch valve and the return valve 1¼-inch. The above sizes were actually installed. The architect did not feel like reducing the sizes when his attention was called to same, saying, "He did not believe the engineer would approve the change." The writer understands the word "Engineering" to mean "Educated common sense applied in straight thinking to material problems."

The writer also calls to mind a 2-inch wet return run underground for 800 feet, and carrying the condensation from 6,000 sq. ft. of radiation surface that is giving the best of satisfaction.

There also appears to be an erroneous idea afloat amongst those that have to do with steam heating that "it takes more radiation at 8 oz. pressure with a "vapor" system than it does with low pressure steam at ½ lb. gauge.

If it is within the province of any living man to explain "why" this should be, the writer would like to be enlightened.

It has become common practice in some sections of the country to add 60% to the amount of radiation found for low pressure steam in order to arrive at the amount required for "vapor." Assuming "vapor" at 8 oz. and steam at ½ lb., both having a temperature of 213.5F, why the 60% addition?

Why any addition unless the radiator is to act as a partial condenser?

These are a few of the fallacies that will tend to prevent the adoption of standards as suggested by the title of Mr. Donnelly's paper.

Mr. Bushnell: Perhaps no paper has been written during the past year which is more interesting to the practical heating engineer than the paper under discussion.

The subject appears to have been treated throughout in a very logical and comprehensive way. Quite a number of empirical formulae have been prepared by various engineers to determine the heat losses in buldings in various localities. All of these formulae if they are at all logical must depend on the basic conditions described in this paper.

As Mr. Donnelly states, the number of square feet of radiation in a building at a given outside temperature is obtained by adding the losses in heat units per hour through the walls and glass surface to the losses in heat units per hour on account of air changes or leakage and then dividing by the amount of heat units per hour given out per square foot of radiation. This would perhaps be well expressed by the basic formula.

$$R = \frac{(G\ lg + W\ lw + V\ lv)}{K}$$

where

R = Square feet of radiation required.

G = Square feet of glass or door openings.

W = Square feet of net wall area.

V = Cubic feet of air changed per hour.

lg = Loss of heat units per square foot per hour through windows.

lw = Loss of heat units per square foot per hour through walls.

lv = Loss of heat units per cubic foot on account of air changes.

K = Heat units given out per square foot per hour by radiation.

To determine the amount of heat required for a building during the heating season, it is, of course, necessary to assume a certain number of hours during which the radiators are heated and giving off heat units.

It is interesting to note that Mr. Donnelly's figures can be checked in a general way by the general experience of heating companies. It is generally conceded that the average number of pounds of steam ,consumed per square foot of radiation per season runs about 700 pounds, although different buildings vary widely from this average figure. This applies to places having an average temperature of 39° Fahrenheit, during the eight months heating season which corresponds to the average temperature of Chicago and vicinity. When the steam is sold on a flat price per season the average is usually somewhat higher than this, and when the steam is sold on a meter basis the average is lower. It is perhaps fair to assume that during the heating season of eight months, the radiation will be turned on, an average of 12 hours per day. There are, of course, times when radiation is required 24 hours per day and other times when it is turned off entirely.

Taking Mr. Donnelly's average heat unit per square foot of radiation per hour of 250, and multiplying this by twelve hours per day, times 240, we have 720,000 B.t.u. given out by radiators during the heating season. Assuming that in round figures every pound of steam will give off about 1,000 B.t.u. we would have 720 pounds per square foot of radiation as the requirements for the heating season.

Another interesting point which is touched on in Mr. Donnelly's paper is the saving which can be made in radiation where the indoor temperature is to be at a lower point than the so-called standard of 70°. He shows that if the room temperature is to be at 65°, the square feet of radiation required is only 89% of that required where the room temperature is to be at 70°; if the room temperature is to be at 60°, the square feet of radiation required is only 78% of that required for keeping the room temperature at 70°. This is based on the requirement during zero temperature.

As a matter of fact the average saving in coal requirements by reducing the room temperature required is greater proportionately than the saving in radiation. Take a locality such as

Chicago or New York where the radiation is figured on the basis of zero temperature, a difference of 10° would make a saving in coal of only 1/7th or about 14% during zero temperature. Under average conditions we have a difference of 70° minus 39° or 31° as the average difference between outdoor and indoor temperature. If by changing our average difference between room temperature and outdoor temperature, from 31° to 21° we would have a saving of more than 30%.

In an address given before the National District Heating Association, last June, Mr. John F. Gilchrist of Chicago estimated the amount of coal burned per season in the United States at over 500,000,000 tons. After deducting the amount burned by the central station lighting companies, the street and electric railway system and the steam railways, there was left a total of about 390,000,000 tons. While it is impossible to get any accurate data as to how this latter amount is divided between manufacturing purposes, cooking purposes, and the heating of interiors, there is no doubt but what a very large portion of the coal is used for interior heating. Assuming the amount of coal used in this way to be 200,000,000 tons and assuming the average cost of the coal (a large part of which is Anthracite and costs from $5.00 to $7.00 per ton) to be $2.50 per ton, we would have a total cost for heating interiors of $500,000,000 per year. Assuming that these interiors are now heated to our present standard of heating, 70°, it is easily seen that the English standard of 60° would mean a saving of about 30% of $500,000,000 or $150,000,000 per year. No wonder the English consider Americans extravagant. There are many, doubtless, who believe that a temperature of 70° in a building is necessary to maintain the health of the occupants. Others believe that it is largely a matter of accustoming oneself to a cooler temperature and wearing suitable winter clothing. A great many medical authorities claim that a large part of the sickness in winter time is due to excessive heat in apartment houses and buildings. A change in the standard of heat might not make a great deal of difference in the amount of radiation to be employed, as we usually find that during zero temperatures, especially when a high wind is blowing, the average interior is not likely to be much over 60°. There would be, however, a very large reduction in the consumption of coal during the heating season and the matter is certainly worthy of serious consideration.

Mr. Donnelly shows that the number of heat units given off by a radiator per degree of difference in temperature increases

about 2% for each 10° increase of difference in temperature. I would like to ask him if this is not due to increase of air circulation around radiators due to increased difference in temperature.

Mr. Cooley: Mr. President, I would like to ask with regard to the Standard Sizes of other return mains. Mr. Donnelly speaks only of the wet returns, or those which are below the water line, and dry returns in which he carries steam. I would like to know if there was any consideration made of the dry returns in the vacuum and vapor systems, in which they are not supposed to carry steam but do have to carry air from the radiators. There is not anything said on that subject.

The Chairman: 'Question number two, Mr. Donnelly. Any further remarks. Mr. Donnelly, will you give us your answer to these questions, please?

Mr. Donnelly: Mr. Bushnell's question as to whether 2% increase in the coefficient is due to increased air movement over the radiator, I hardly think I can answer. I have had a suspicion that it is due to increased radiation activity, that is that the radiated heat given off increases as the temperature goes up, much more than the heat given off by convection. However, I have never gone into the subject very far. I have not attempted to find a reason for it, but to use the coefficients as they are given. I do not know that I can give any other reason. Mr. Cooley's question as to the sizes of returns for vapor and vacuum systems that are not supposed to carry steam, so far as I know no one has ever published anything on return pipe sizes that has taken that into consideration. All the manufacturers of appliances and apparatus that keeps steam out of the returns are so extremely modest, that they have never said anything about it (laughter). There are a number of them that I know, who know a great deal about it, but they must be like the Trade's Union man which I referred to, and I assure you in a very good-natured way, and not wishing to criticise that body unduly, because they do not want to tell, because the other man would then know. If I knew, I would tell, that is one of my failings.

In regard to what plants do after they are installed; as contractors and engineers, we do not follow this investigation for as many years as the National District Heating men do, hence their view is entirely different. They want to know what a plant will be doing ten years from now, when it gets its full quota of customers, especially if it has been paying 20% for a few years

as we hope all the National District plants will do for some time to come.

Mr. Donnelly: Mr. Tait speaks of the impossibility or the very great improbability of establishing a standard. I do not think I will take the time to go through it now as the hour is late, but just in line with the remarks I gave before I read the paper, the National Heating people have rushed in, being commercial men, and wanting to adopt standards, where the engineer won't follow, and if the engineer is going to back out, perhaps we will have to keep it in the National District Association. They will rush in as long as their work is appreciated. I might say the Committee got together and considered different things, got suggestions from outside, and assigned different subjects to different people and that is why this subject was assigned to me by the Chairman, and in like manner a subject was assigned to Mr. Bolton, and several subjects to other members of the Committee. I think that may be a good way, possibly, to get the Committee's work done.

Mr. Tait speaks of 2-inch pipes, for instance, of a wet return or draining 6,000 square feet. I do not think that is doing very well. Some years ago if you had looked over a paper of mine, you would have found that I had rated a 2-inch wet return pipe as capable of draining 12,000 feet, rather than 6,000.

I tested a plant in New York, with 12,000 square feet and there was only a 2-inch pipe back to the boiler. All the water ran back nicely. Mr. Lewis of the Society stated that he had also modified his practice and cut down the size, which had formerly been 4 or 5 inches, dropping that down first to 3 inches, and afterwards to a 2 inch, as a wet return, and had found that 12,000 feet of radiation condensation could be carried back through a 2-inch pipe to the boiler without trouble.

Mr. Cooley: There is one matter more I would like Mr. Donnelly to answer, that was in regard to the sizes of the valves, the dimensions he gives for vapor work. Mr. Tait brought that up.

Mr. Donnelly: With regard to that table of pipe sizes, I submitted it to a few men whom I consider engineers, before I set it up, and it is surprising how quickly some specifications came out with those sizes in them, and I am going to say to the heating and ventilating engineers to-day that if they are not very careful, that table will be a standard, and if you get a table out with any indication that it might possibly be near the standard, if it once gets to use, it is very difficult to kill it after-

wards, so that if there is anything wrong about it we should have a volume of protests. I was in a contractor's office a little while afterwards and he said, "Mr. Donnelly, I have had a number of men in here who all say that half an inch return valve will successfully drain one hundred and fifty square feet, but you and your trade publications say forty square feet, and this specification which we have from this architect's office, they say forty square feet." Well, I said, "I submitted that table to this architect's office, and that engineer, and he said that agreed with their practice," and it must have agreed with their practice, because he has put it in his specifications.

Some years ago I was Chairman of the Committee of the Society of Heating and Ventilating Engineers to establish a standard of steam and return pipe sizes. Mr. Hugh J. Barron and Mr. Frank McCann of New York were on the Committee with me. I sent letters to 150 or 200 or even 300 members and got their practice, and we adopted or recommended one ounce drop to a one hundred foot straight pipe as a standard.

Professor Kent read a paper and recommended a drop of one-tenth of one ounce to one hundred feet in pressure. I really believe that one ounce to the one hundred feet is very close to the standard, or average practice of the profession as it stands to-day.

Mr. Jellett: I will give the reasons why the branch return from the radiators are as a rule very much larger than they are needed, for the physical reason that people will step on them, and bend them out of shape. I put large sizes in for protection only. I have reduced the size, but when they are exposed above a floor, from a radiator, men will step on them, and if it is a small pipe it may be broken and bent out of shape.

CCCLXXXIX.

THE ESTABLISHMENT OF A STANDARD FOR TRANSMISSION LOSSES FROM BUILDINGS OF ALL CONSTRUCTIONS*

BY REGINALD PELHAM BOLTON

At first sight this subject appears to be, at the present time, somewhat elementary. Common practice has accepted results of certain experimental tests as determining the transmission of heat through building materials.

The pioneer work of Peclet in this direction has been followed by experiments conducted under the auspices of the German and French governments, and some individual work has been effected by other observers. The subject has long been presented in standard works such as those of Box and Carpenter, and in technical papers and pamphlets by Doctors Kenealy, Allen and Carpenter, by Messrs. Hubbard, Hogan, Wolff and Harding.

But, as Dr. Jas. Hoffman says, "many of the values are only rough approximations at best." The various test results were compared by this author, who found quite a divergence between the references and prepared a table representing "a fair average" of all of them. These disagreements indicate that the matter is by no means conclusively decided. There are, to begin with, great variations in the character of the materials which are classed under a single description.

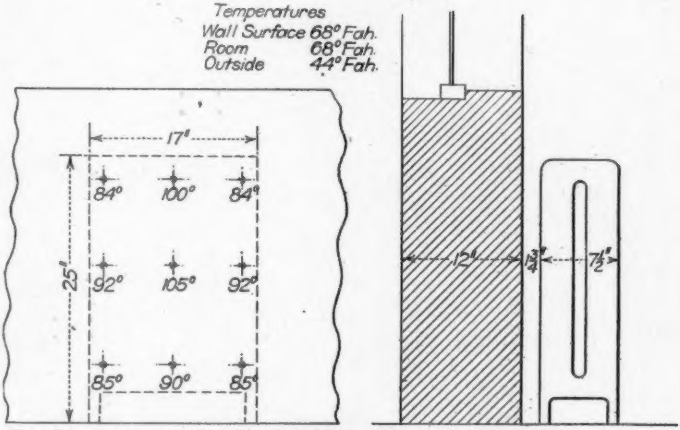
A "brick wall" may be of very variable materials in different localities, and "frame construction" may vary very largely in form as well as in workmanship.

But apart from these explainable divergencies, there is by no means a positive determination as to whether the *rate* of transmission is the same for a small difference as for a large difference of temperatures. This is a fundamental element which calls for definite determination. The rate of heat transmission should,

* This paper is part of a report of the joint educational committee of the National District Heating Association and this Society read at their annual convention in Chicago, June, 1915.

theoretically, be in proportion to the difference of temperature between one surface and another, and this is the generally accepted assumption.

It may be nearly or wholly correct for materials of a homogeneous character such as metal, but there is at least a probability that the rate may vary through composite materials, and through those offering a high extent of resistance to heat transfer.



The figures in area enclosed by dotted lines show temperatures in degrees Fahrenheit of the wall behind radiator

In such materials it would seem possible that the rate of transfer would fall off materially at low differences between opposite sides of the material. If this is the case, it may account for some of the divergence between results observed in different experiments. This part of the subject is therefore still open to further investigation.

There are three forms of heat transfer through building materials, viz.: radiation, convection and conduction. The common acceptance of the term Radiator or of the action of Radiation as applied to heat transfer, needs correction. The term is a misnomer as applied to heating apparatus commonly known as a "Radiator," since the heat *radiated* from such low temperature surfaces is a minor part of the work of heat transfer which they effect. They should be more properly termed "Convectors" since by means of air-motion over their surfaces and physical transmission of heat by contact with it, they effect the major part of their work.

Available information upon the subject of radiated heat, its laws, its methods and its comparative effects is extremely deficient.

We hardly know whether the transference of energy which takes this form is really heat or is heat transformed to energy, which is retransformed to heat in the body or surface upon which it impinges or through which it passes. But whatever be its character or extent, such as it is, it is largely wasted in our present methods of installing heating apparatus. The modern method is to place "radiators" backing on an exterior wall, generally under a window.

Whatever be the effect of radiation, one-half of the heating element from which it is derived directs the rays to this wall, and not to the interior space required to be heated. The wasteful effect is increased by the action of convection, passing a column of warmed air between the "radiator" and the wall. The result is that the wall surface behind each radiator becomes heated to a temperature far in excess of that of the room or interior space and the transmission of heat through that part of the building construction or material is substantially accelerated.

Observations upon such an instance as that illustrated here, showed an average temperature of the wall surface of 90.7 degrees, nearly doubling the difference between the mean exterior temperature and that part of the interior surface of the wall.

SUMMARY OF OBSERVATIONS UPON TEMPERATURES OF WALL SURFACES BEHIND RADIATORS

Average seasonal difference—Interior of room to exterior—70-44	26 degrees
Average temperature of wall behind radiator	90.7 degrees F.
Average temperature of wall surface of room	68 degrees F.
Heat transmission per hour through wall behind radiator at .33 h.u. per square foot per degree difference in temperature	15.4 h.u. sq. ft.
Heat transmission per hour through wall surface other than behind radiators (68-44) equal 24 degrees difference at .33 h.u. per sq. ft. per degree difference in temperature	7.92 h.u. sq. ft.

The rate of heat transmission per sq. ft. through the walls behind radiators is approximately twice the rate through the other exposed wall surfaces of the room.

The second process of heat transfer is that with which the study of the transmission losses is chiefly concerned. The process of conduction is a flow of heat from one part to another

physically communicated. It is probably accelerated, as the foregoing enquiry tends to show, by radiated energy passing into, and perhaps through the materials.

CONDUCTIVITY OF VARIOUS MATERIALS ENTERING INTO THE EXTERIOR CONSTRUCTION OF BUILDINGS AS DETERMINED BY VARIOUS AUTHORITIES AND AUTHORS.

Materials.		*Heat units per degree difference per square foot,*							
Substance	Construction	Peclet (Carpenter)	German Government	Austrian Government (Macon)	Greene 1913	Recknagel and Reitschel	Carlton F. Tweed	Harding	Hoffman
Glass	single window	According to height (see note)	1.09	1.07	0.96	1.03			1.0
	single window (wet)				1.1				
	single window (wire glass)			1.04			1.35		
	double windows		0.418	0.47	0.41	.472	0.50		0.6
	single skylight		1.118	1.14	1.06	1.090	1.10		1.1
	double skylight		0.621	0.48	0.51	.492	0.60		0.7
Brick	wall, 8 in. thick	0.37	0.46		0.39	.37	0.39		0.4
	wall, 12 in. thick	0.32	0.32		0.31	.29	0.31		0.31
	wall, 16 in. thick	0.28	0.26		0.25	.25	0.25		0.26
	wall, 20 in. thick	0.25	0.23		0.21	22	0.21		0.23
	wall, 24 in. thick	0.24	0.20		0.18	.19	0.18		0.21
	wall, 28 in. thick	0.22	0.174		0.16	.16	0.16		0.19
	wall, 32 in. thick	0.21	0.15		0.15	.14	0.15		0.17
Brick and Plaster	B. & P. 8½ in. thick			0.43	0.37	.36		0.5	
	B. & P. 13 in. thick			0.34	0.29	.28		0.33	
	B. & P. 17½ in. thick			0.29	0.24	.24		0.25	
	B. & P. 22 in. thick			0.24	0.21	.21		0.21	
	B. & P. 27½ in. thick			0.21	0.18	.18		0.17	
Brick furred and plastered	B. F. & P. 8½ in. thich				0.24		0.24	0.41	0.28
	B. F. & P. 13 in. thick				0.20		0.20	0.3	0.217
	B. F. & P. 17½ in. thick				0.18		0.18	0.23	0.18
	B. F. & P. 22 in. thick				0.16		0.16	0.2	0.161
	B. F. & P. 26½ in. thick				0.14		0.14	0.16	0.147
Limestone	wall, 16 in. thick			0.56		0.43			0.39
	wall, 20 in. thick			0.50		0.38			0.35
	wall, 24 in. thick			0.45		0.35			0.31
	wall, 28 in. thick			0.41		0.31			0.28
	wall, 32 in. thick			0.36		0.28			0.25
Sandstone	wall, 12 in. thick			0.58		0.45	0.28		
	wall, 16 in. thick			0.51		0.39			
	wall, 20 in. thick			0.45		0.35			

Sandstone	wall, 24 in. thick			0.41		0.31	0.23		
	wall, 28 in. thick			0.37		0.28			
	wall, 32 in. thick			0.31		0.26	0.19		
Concrete	wall, 4 in. thick							1.07	
	wall, 6 in. thick							0.7	
	wall, 8 in. thick			0.49				0.53	
	wall, 12 in. thick			0.43	0.48			0.36	
	wall, 16 in. thick			0.37				0.26	
Wood	clapboards							0.47	.3 average
	clapboards, paper lined							0.34	
	clapboards, sheathed				0.31			0.3	
	clapboards, sheathed and paper lined							0.26	
	ditto, and brick plastered				0.20		0.25	0.21 Donnelly	
	concrete, cinder fill, 4 in.						0.6		
	concrete, cinger fill, 6 in.						0.5	0.43	
	slate sheathed			0.43			0.43	0.36	
	tin sheateed							0.31	
	shingle sheathed							0.29	
	tar and gravel							0.29	
	mill tar and gravel							0.2	
	shingle roof sheathed				0.31				
Doors			0.414		0.34		0.31		
					0.55	.410	0.48		
	hardwood, ¾ in.			0.69					
	hardwood, 1 in.			0.63					
	hardwood, 1½ in.			0.53					
	hardwood, 2 in.			0.46					

NOTE.—Peclet gives rates of heat transmission through glass, according to heights, as follows:

Window Height	Transmission in B.T.U.'s per sq. ft. per degree difference per hour
3 4t. 3 in.	0.98
6 ft. 7 in.	0.945
10 ft. 0 in.	0.93
13 ft. 3 in.	0.92
16 ft. 3 in.	0.91

—Compiled by Reginald Pelham Bolton. April 1915.

But the process of conduction in building materials evidently varies by several conditions of the structure affected. The solidity of the mass is one. A brick of a spongy character, having many air particles in its composition must resist the conduction of heat much more than one of semi-metallic nature, or of densely compressed material. So that "bricks" should be further defined, and we need to learn something more of the heat-conducting qualities of them and of other materials such as woods, commonly used in buildings, before a standard can be set for the rate of heat conduction.

Perhaps, even more than the relative homogeneity of materials, their heat carrying capacity is affected by moisture. Water is a good conductor of heat compared to air, or to calcined substances; so that if a wall is water-soaked, its conductivity is thereby increased. Moreover, if its surfaces be wet, the transfer of heat from the surface to the air moving over the surface is bound to be accelerated.

It must not be forgotten that, in assuming a rate of heat transfer *through* building materials, the agent which imparts, and the agent which conveys away that heat may play a most important part. Hot moist air inside, with a damp wall surface should be an excellent agent to impart heat to a wall. So also would be the effect in removal of heat of a wet wall and dry air outside.

Finally, the third element, convection, plays a large part in the rate of transmission of heat through wall and building surfaces. "Still air," inside or outside a building does not imply that even in the entire absence of wind movement, there is no movement of air upon the respective surfaces. Under such conditions an active motion is proceeding over both the surfaces, due to the acceleration of air movement by the dissipation of heat on the one side, resulting in a falling air current, and the reception of heat on the other, resulting in an upward air current.

The rate of these two motions must affect the rate of conduction of heat through the wall or other material. Such effects are well established in the process of heating water and in condensing steam. In such apparatus water is forced at high speed over heated surfaces greatly accelerating the heat transfer. We must suppose that very similar effects follow the action of quick air motion over building exteriors. But even with no wind, there is a lively upward motion of the enveloping layer of air around a building. The whole of the heat generated in a building has to find its way out through its surfaces, partly by leakage, but mainly by conduction. Every building therefore may be con-

sidered as a sort of "pyre," up the sides of which a column of heated vapor is whirling, rolling over on itself and joining a volume ascending from its windows and roof, which in still air ascends to some height before being fully absorbed into the aqueous atmosphere.

And it is not to be forgotten that each building may also be a large, though feeble, "radiator" directing low temperature rays of energy through surrounding spaces or into contiguous buildings.

From these considerations, it would seem that a vast amount of research, experiment and comparison remains to be done to decide the exact conditions of heat conduction through buildings, before the subject can be regarded as satisfactorily standardizable.

But as, in the meantime, and while this fascinating subject engages the further attention of the members of the heating profession, there must be some practical, approximate, or average data rendered available for our current purposes, it may be of service, to gather for our records and reference, a compilation of the published results, which are, therefore, here presented for observation and use as judgment and experience may dictate.

Thoughtful consideration of some of the contingent elements to which attention has here been drawn, will lead the student to apply to the use of these approximate determinations, a cautious liberality by providing ample means of heating to cover possible deficiencies in the determinations of the rate of heat transfer through building materials.

ABSTRACT OF PAPER

The paper quotes the various authorities to determine the heat losses of different building materials and gives tables showing their values.

Explains the texture of different materials and how one differs from the other in transmitting heat through building walls.

Explains how the heat is lost by the movement of air in contact with the surface of the building.

DISCUSSION

Mr. Donnelly: Mr. President, I did not prepare a discussion on this paper, and I do not know if I am capable of preparing any discussion, but there are some things that I think I can discuss a little.

I am quite interested in Mr. Bolton's taking the temperatures behind the radiators, and figuring out the increased loss by reason of the higher temperature of the wall. It would be probably very difficult to get the increased loss due to the higher temperature of the air passing the window above the radiator. No doubt a thermometer could be hung fairly close to the glass, and thus get some idea of the temperature of the air, leaving the radiator as it passes up in front of the glass, and also to see how high and how close to the glass it goes. Naturally there might be a descending current of cool air between the warm air and the glass, and then, of course, the leakage of the window would affect this problem very much.

Referring to the tables of transmission, there are really two problems. One reason Mr. Bolton speaks of, and another that I call attention to. One is as to whether the transmission is proportionate to the difference in temperature between the inside air and the outside air; and the other is whether the transfer is proportionate to the thickness of the wall. Mr. Harding of the State College at Pennsylvania has made a good many tests of insulating material for refrigeration, and I believe refrigerating engineers consider the transmission through the walls of the refrigerator or cold storage room, is proportionate to the thickness. In fact, their tables I think are usually given as to the rate of transfer per inch of thickness, and if they wish to cut the transfer in half, they made the wall double thickness.

It is interesting to note that all the older tables of transmission, through walls, are not proportional with the thickness. If you look under the authorities who have written on the subject, and note a brick wall 12 inches thick, and a brick wall 24 inches thick, you will see that they agree in the main, that twice the thickness of the wall is about two-thirds the transmission, instead of half the transmission, as it would be if the transmission was unevenly proportionate to the thickness.

We have no figures from Mr. Harding on a brick wall, but we have it on brick and plaster. If you will note opposite brick and plaster, 13 inches thick, of Harding's experiment, it gives .33, and opposite 27½ inches thick, it is approximately one-half or .17. On the next page, on the concrete walls, you will note opposite 8 inches thick he gives .53, and opposite 16 inches thick, .26, just about one-half the transmission. Another authority on concrete walls the Austrian Government, translated by Macon, gives concrete wall 8 inches thick, .49, concrete wall 16 inches thick, .37.

I rather think Harding is correct. I think those other allowances were made by reason of the absorption of heat by thicker walls. You will find usually that buildings with very thick brick and stone walls are not continually occupied as a usual thing, such as warehouses or heavy factory buildings. They are allowed to cool down at night, and in the morning the walls absorb a great deal of heat and that is one phase of the necessity for the application of heat in bringing a building up to the temperature from which it has originally been dropped. It would, of course, be necessary to find the specific heat of the building, which would be a very difficult thing. If we put a recording thermometer in a room which has been allowed to cool down, and note the rate at which it takes on heat, and did that in buildings of different types and different constructions, you will find that under similar conditions one building would take on heat faster than the other, because the specific heat of one building would be much lower than that of another.

No doubt the specific heat of brick is very much less than the specific heat of stone, and in the same way the specific heat of a frame construction building is less than the specific heat of a brick building, therefore it would go up in temperature faster, and the building itself would not require so much heat. It is, of course, well known that a building in which a great deal of iron or metal is stored, has a very high specific heat, so that it is very uneconomical to allow a building in which much metal is stored, or in which water, or tanks containing liquids is stored, to allow them to cool down very far, because they would absorb a very great deal of heat in the materials stored in the room, in bringing up the temperature again. It is for that reason that machine shops which manufacture heavy machinery, bolt, and wire, and screw machinery, should be kept warm over night, and over Sundays and holidays.

Therefore, I think, in modifying these transmissions for thicker walls, has been largely influenced by the fact that thicker walls require more radiation, and, therefore, they must have assumed that the transmission was greater, while it may have been the absorption of the wall. Harding's experiments in some former transactions are therefore worth looking at. He had a box in which the temperature was raised by electrical resistance coils, and measured the heat given off by the electrical current, so that there was not much room for error, and it seems that he really could determine that point as to whether the heat transfer was proportionate to the thickness. Mr. Bolton speaks of damp

walls. I do not think he refers directly to the evaporation of water from the walls, which, of course, would mean a great deal more heat required, and perhaps a tabulation of more transfer of heat, though the transfer would, of course, drop down if the wall ever became dried out. Naturally, brick walls being greater absorbents of rain and moisture than stone walls, would apparently have a greater transmission when the walls were wet; whereas the heat would not be heat of transmission, but a heat used for evaporation.

I think that Mr. Bolton, when speaking of a building as a radiator giving off this heat to the outside atmosphere, gives a good description of the loss of heat from a building. I remember standing in a Boston street one cold and stormy night and a man who was with me said that he could feel the heat radiated from the building. He wanted me to see if I could observe it too. I do not know that I could, but he said that he could feel the reat radiated from the building.

Mr. Cooley: There is one point about this first chart on page 31 that I do not believe is made enough of, as to the effect of the higher temperature between the radiator and the wall having an effect on the rate of transfer of heat from the radiator. In this case, where the temperature outside was something like 40 or 44 degrees, they obtained a temperature of 90 behind the radiator. Take the outside temperature, zero, and you would not have as high a temperature behind the radiator as there was in the case noted, and the radiator would oe more efficient under those conditions. Also there would be less difference between the amount of heat radiated through the wall behind the radiator, and the amount radiated through the wall connected with other parts of the room. I have found in several cases, in actual practice, that would actually bear out that statement. I have seen cases where 160 degrees of water temperature was just sufficient to heat the rooms properly, when the outside temperature was at 40, and had the outside temperature drop to 20, and did not raise the water temperature more than 5 degrees, but was successful in heating the room, but one had to burn a great deal more coal, indicating that the lower outside temperature had an effect upon the heating efficiency of the radiator. The difference was between the inside and the outside temperatures, as the room temperature was exactly the same, and I believe that more experiments along that line ought to be taken up. We have, all of us, assumed that the amount of radiation required with a minimum temperature of 35 was one-half of

that required if the temperature outside was at zero, but from Mr. Bolton's experiments, and from actual practice and observation, it looks to me that it would not be so; we would have to have more than half as much radiation as was required for zero. I think that an experiment along this line could be conducted in some University where they have facilities to do that. I would suggest tests using water radiators, and that the water be heated in a vessel electrically, so that they would know how much heat was imparted to that water, and thus a careful record could be attained to determine what effect variation in the efficiency of that radiator was, under different conditions. This will affect us in another way, that is in our rules that we have for testing a plant designed to heat to 70 degrees in zero weather, when the outside temperature is higher. It has always been assumed in carrying out these rules that the variation in heat transmission was proportional to the difference in the temperature, if it is not so, the heating system is getting the short end of it. I would like to see the fact tried out experimentally by someone who is well equipped to make the experiment.

The President: Any further remarks on this paper? I think this field needs a great deal of study and investigation. One of the leading investigators of ventilation in Providence, talking to me some time ago, remarked that he did not know any field problem that needed more study and investigation than that. Mr. Donnelly had spoken of Professor Harding's experiments, and I believe Mr. Blackmore told me that Professor Verner is Chairman of the Committee investigating that subject, and is at work on the problem.

CCCXC.

AN EXPERIMENT IN VENTILATING A SCHOOL ROOM*

One of the first experiments determined upon by the Chicago Commission on Ventilation pertained to the ventilation of a schoolroom. For a long time many of the teachers of the public schools of Chicago complained of the ventilation within their rooms. Therefore, with the hearty approval and co-operation of the Board of Education, the experimental work was undertaken in the autumn of 1910. The attitude of the Board of Education has always been to improve the present system of mechanical ventilation within our public schools, if possible. All expenses in connection with the experiments on schoolroom ventilation are gladly borne by the Board of Education.

The public school buildings of Chicago are equipped with the plenum system, operating at a pressure of approximately one-half ounce.

The Chicago Normal College, Sixty-eighth Street and Stewart Avenue, was decided on as the building in which to make the experiment.

The experimental work has continued from the autumn of 1910.

The first tests made were somewhat in the nature of checking up the work of the builders. A rule of the Board of Education requires the delivering of a minimum of 1,800 cubic feet of air per pupil per hour. Anemometer readings made in several of the rooms of the Normal College showed that this amount was being delivered. There remained, however, a closely related question: namely, whether or not each pupil is supplied with his share.

Studies were made for air currents in two classrooms and one laboratory room.

*Reproduced by permission of the Chicago Commission on Ventilation. The Society takes this opportunity of recommending to the members the purchase of a copy of the complete report which may be obtained for twenty-five cents from the Secretary of the Commission, J. W. Shepherd, Chicago, Ill.

The tests for air currents were made by the use of toy balloons which were inflated with hydrogen gas and counterpoised in the rooms by means of improvised weights. In addition to the toy balloons, small turbine wheels were used. These were made from aluminum, cork, and steel needles, and were especially constructed for these tests. The blades of the turbines were made from aluminum and set into hubs of cork. Across one end of a cork hub and parallel with the plane of the blades, was fastened a strip of aluminum containing a slight indentation in which the pivot of the device turned. The fine point of a steel needle served as a pivot, and when ready for use the turbine revolved in a horizontal plane. These turbines are very sensitive to vertical currents of air—in fact, they respond to convection currents from the heat of one's hand. The counterpoised balloons were useful in tracing all air currents, irrespective of their direction, whereas the turbine wheels could be used only in testing for vertical currents.

One of the classrooms tested is 25 by 25 feet and has an east exposure. The other walls of the room have no immediate contact with the outdoors. The inlet and outlet ducts in this room are installed in the north wall, and the air enters the room with a velocity of about 650 feet per minute. When balloons were pushed into the entering currents, they were hurried across the room near the ceiling to the south wall. From the ceiling at the south wall, the balloons usually took one of two general courses, depending largely upon outdoor weather conditions. If the outdoor temperature was low and the wind was blowing directly against the windows, then the balloons moved over to the outside wall, down the wall or windows, and over to the outlet duct. If the outside temperature was moderate, then instead of the balloons crossing over to the outside wall, they were likely to poise in the southeast corner of the room, or possibly move vertically down along the wall opposite the inlet duct to within a foot or two of the floor, and then over to the outlet duct. It was very noticeable that air currents established themselves in aisles and other open spaces along the floor. During the winter season the turbine wheels, when placed on the window ledges, revolved almost all the time. Their direction of rotation indicated the downward movement of a sheet of cold air; moreover, this sheet of cold air was very perceptible to any one seated near the outside wall or windows.

The other classroom is 26 by 45 feet; is a northwest corner room. It has about twice as much exposure on the north as on

the west side. As already stated, there are two inlets and two outlets in this room, and they are located in the long inside wall. The velocity of the incoming current of air was practically the same as in the smaller room. Anemometer readings showed an adequate supply of air for good ventilation. Balloons placed in the incoming current were hurried across the room to the opposite wall, the outside wall with the north exposure. After reaching the outside wall, they almost always went vertically downward to within a foot or two of the floor; then they moved over to the outlets near the floor and almost directly under the inlets. The hot incoming air driven against the cold outside north wall and windows, reduced the influence of wall and window chill. But in cold weather, especially with a north or northwest wind, a downward moving sheet of cold air was very noticeable. The small turbine wheels revolved constantly in cold weather when placed upon the window ledges or near the exposed walls. In the central part of the room, that is, between the two sets of inlet and outlet ducts, the balloons did not show perceptible air currents. Furthermore, there seemed to be eddies in close proximity to the currents at either end of the room.

The laboratory room in which the tests for air currents was made is 26 by 45 feet, and is a northeast corner room. It has about twice as much exposure on the north as on the east side. The room has two inlets and two outlets on the long inside wall. The inlet duct at the west end of the room is so close to the west wall and ceiling that when the current of air is delivered against the north or outside wall, it takes a diagonal course downward and inward along the north wall, and across the windows. By this action of the air current shown by balloon tests, there is a considerable section of the west end of the room in which no perceptible air currents can be found. At the east end of the room, the air current behaves as might be expected, namely, the incoming air crosses from the south to the north end of the room along the ceiling, strikes the cold outside wall and windows, drops vertically downward, and moves over to the outlet. Turbine wheels placed near the windows or outside wall during cold weather were constantly rotating. The direction of rotation indicated a downward current of air.

From the evidence obtained in the use of the balloons and turbine wheels, it is fair to conclude that air within the schoolrooms tested did not act as anticipated in the plenum system. Moreover, the action of the balloons and turbine wheels under varying weather conditions, especially as regards direction and

velocity of winds, warrants the conclusion that in the type of construction with which we were concerned, air distribution is seriously interfered with by conditions for which the plenum system, as generally installed, is not responsible.

The first constructive effort in the ventilation of a schoolroom in which the plenum system had been installed was made in one of the regular schoolrooms of the Chicago Normal College. The schoolroom was fitted up as an experimental room. One of the factors considered in the selection of the experimental room was that of subjecting it to our northwest winter winds.

The experimental room is 24 by 32 feet, on the basement floor, and has a west exposure. The room originally had a thirteen-foot

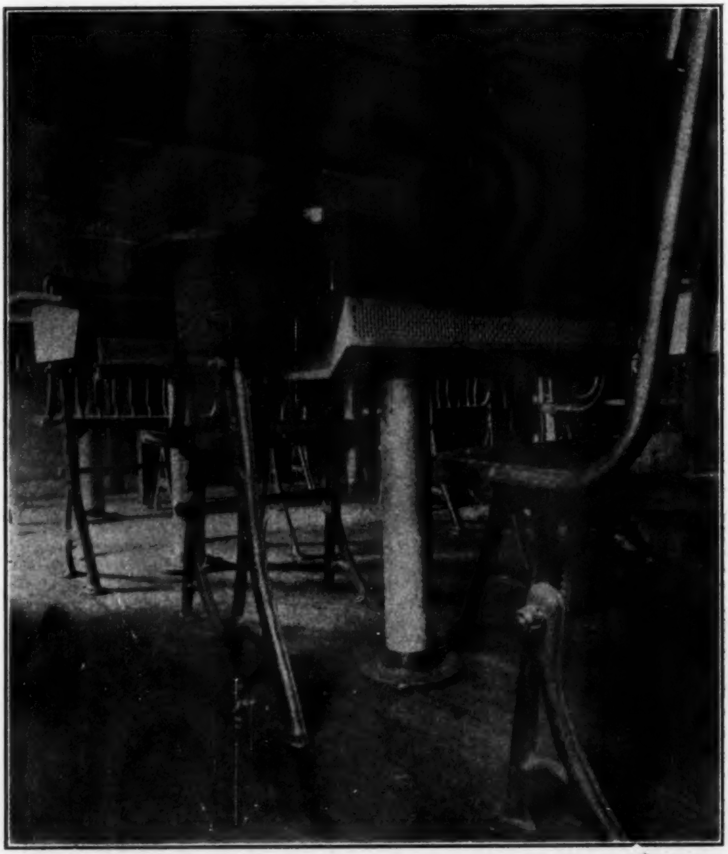


Fig. 1

ceiling. In the original installation for ventilation, air entered the room near the ceiling at the center of the east wall. The main air current was across the room from the east wall to the west (outside) wall, then down the cold outside wall, and back to the outlet duct near the floor in the east wall. The changes made in the room were as follows: First, the outlet duct, near the floor, was closed; then an airtight false floor was built about eighteen inches above the regular floor of the room, and a false ceiling was hung about eight inches below the room ceiling; then an air shaft was constructed to connect the inlet duct of the original installation with the air resorvoir between the floors. See plates 1 and 2.

Outlet duct was tapped near the ceiling connecting it with the compartment between the ceiling and the false ceiling. Three-inch circular holes were cut through the false floor, and galvanized iron pipes, fitted into these openings, led under each desk to within an inch of the desk bottom. Openings also were made through the false ceiling so that air delivered into the room might move on through it. It will be noted that these changes turned the operation of the plenum system upside down. Instead of the air entering at the ceiling and leaving near the floor line, this new scheme delivered the air below the floor, and outgoing currents left the room at the ceiling. As already intimated, the

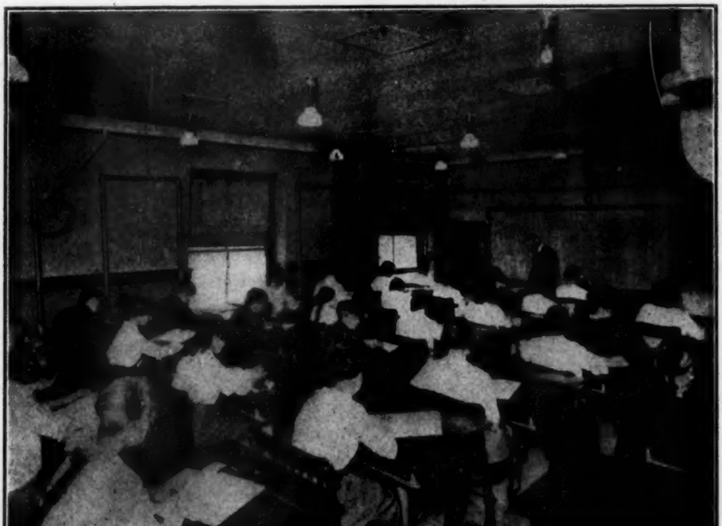


Fig. 2

idea in this scheme was to furnish a positive distribution of air to all the pupils within a room, and also to take advantage of the heat liberated by them in the production of upward moving currents. The new installation was tested in two ways: (1) Simple tests were made with anemometers placed at the edges of the desks. Every test showed an up current. (2) A more striking test than the one with the anemometer, and one as fully convincing, or even more so, was a chemical test made with ammonia and an indicator known as phenolphthalein. This chemical test was made as follows: Linen strings were stretched over the rows of desks at the height of the breathing zone for children seated at the desks. Upon these strings, at intervals of ten or twelve inches, were hung pieces of unsized paper made wet with an alcoholic solution of phenolphthalein. When the room was thus dotted over with these wet papers, it looked much like a laundry drying room flecked with white. Before the wet papers had time to dry, a handkerchief, made thoroughly wet with concentrated ammonia water, was hung in the air duct leading from the plenum chamber, or distributing room, to the experimental room. Within two minutes after hanging the handkerchief in the duct, every paper on the linen threads in the experimental room became red in color. When ammonia is added to a colorless solution of phenolphthalein, the solution becomes red; therefore, the change in the color of the papers was conclusive evidence that ammonia from the handkerchief had been distributed to every piece of paper wet with the alcoholic solution. The test was repeated at another time with the same result. Still another test was made which contained an added feature. In addition to the papers suspended in the breathing zone over the desks and seats, others were hung on strings stretched parallel with those over the rows of desks about seven feet from the floor, but directly over the aisles. In this test all the lower papers reddened in approximately the same time as before, and the upper ones reddened soon thereafter. These tests are conclusive evidence that the air in the experimental room is delivered to each desk, and that the movement of the air in the room is upward, and quite uniformly so. Moreover, anemometer tests made at the outlets of the galvanized iron tubes before the desks were placed, showed that each tube delivers approximately the same volume of air in unit time.

As soon as the before-mentioned changes in the experimental room had been made and tested, the room became a regular high-school room, in use throughout the day. It is customary

for classes to change rooms for different recitations, and, therefore, the experimental room was occupied by different classes each hour. This arrangement added somewhat to the difficulty of our experiments.

However satisfactory the quantity of air furnished for the ventilation of a room, and however satisfactory may be the means employed by properly distributing it, both of which in the long run are very important, nevertheless the human body makes an immediate demand which may overshadow either or both. Immediate physical comfort is the standard of the human body, whatever the consequences, as exemplified either in the drowsy stupor that descends on one immersed in a hot, stifling atmosphere on a cold wintry night, or in the quiet repose that comes from a balmy summer breeze outdoors. Good ventilation shall produce immediate comfort.

One of the most prominent as well as immediate factors in the production of comfort, is temperature, and therefore a study was made to determine the best temperature for a schoolroom. The comfort of the human body is largely influenced by the temperature of the surrounding air, and also, and at the same time, by the rate at which perspiration may evaporate into the air from the body. Relative humidity influences the rate at which such evaporation occurs, but it is only in recent years that much consideration has been given to atmospheric humidity in relation to temperature and comfort.

It has become traditional in this country that the best temperature to maintain in a room is 68 to 70 degrees. There are, however, some who urge that these temperatures are too high, and they cite the English practice of 59 to 62 degrees as evidence of their claim. The difficulty with both these positions is that in deciding on the best temperature, proper consideration is not given to relative humidity. Any adult knows that sultry days are much less comfortable than days of even higher temperature when the atmosphere is comparatively dry. This well-known fact of outdoor experience must be taken into account, especially since it is now recognized that in cold weather we need to humidify air indoors. On this point of humidity, it may be said that the human organism seems to be adapted to a large range of relative humidity, but it is not accustomed to abrupt changes such as one might experience on a cold day in passing from the outdoors into a heated room. In a word, it seems important from the standpoint of health and comfort to maintain a fair

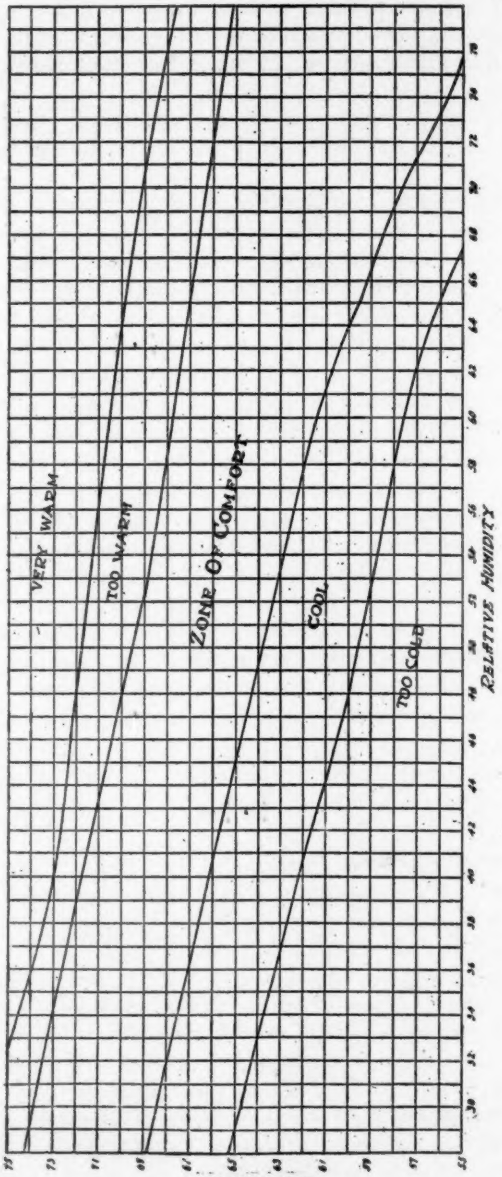
degree of correspondence between the relative humidity of outdoors and indoors.

Any system of ventilation, to be practicable, must produce a feeling of comfort, and therefore both the temperature and the relative humidity of the air are important in ventilation. Temperature and relative humidity jointly help to determine comfort.

It has generally been considered that a temperature of from 68 to 70 degrees with a relative humidity of 70 per cent., is a most desirable condition to obtain (the 70 per cent. relative humidity also is largely traditional). In our tests it was assumed that the best temperature may or may not be 68 to 70 degrees; and also the most satisfactory relative humidity may or may not be 70 per cent.

The experimental room was equipped with an automatic temperature control, and also an automatic humidity control. Moreover, the temperature in the different parts of the room was determined by means of a sling psychrometer. For the most part, the tests on relative humidity and temperature in relation to comfort, were made by a member of the Commission and a graduate student from the University of Chicago. Frequently individual high school pupils in the room were asked whether or not they felt comfortable, and in each case the pupil answering did not know that any other pupil had been asked. The teachers in charge of the room also were asked for opinions. All these opinions, together with our own, served as a basis for record.

Before working very long, it became evident that there was a temperature and humidity range within which the occupants of the rooms were comfortable, and this range gave rise to what is called the Comfort Zone, Fig. 3. This term, comfort zone, means that there is a maximum temperature with a minimum relative humidity. Under the conditions with which we were working, the relative humidity between which limits the occupants of a room are comfortable. In other words, there seems to be no best temperature and also no best relative humidity; but the maximum temperature at which one is comfortable will be associated with a minimum relative humidity, and the minimum temperature for comfort will have associated with it a maximum relative humidity.. Under the conditions with which we were working, we found that a temperature of 64 to 70 degrees with a corresponding relative humidity of 55 to 30 per cent., seems to be the limits; that is, the comfort zone for us was between 64 degrees and 55 per cent. and 70 degrees and 30 per cent.



TEMPERATURE

Fig. 3

It is worthy of note that with a temperature below 67 or 68, with a proper relative humidity, the pupils were better able to give attention to their work than if the conditions were otherwise.

The problem of how to prevent outside wall and window chill from seriously interfering with ventilation, has never been satisfactorily solved.

With the change in installation in the experimental room, it was necessary to make some provision for preventing a sheet of cold air from falling down the exposed walls and windows. We tried to do this by installing steam pipes along the window casings just in front of the windows. The idea in this installation was to induce convection currents around the windows, and in this way prevent the downward currents of cold air. Tests were made of the scheme in two ways: (1) By the use of small turbines it was found that down currents of air were established from a few inches to a foot or more above the horizontal steam pipes. A second method was that of getting temperature readings at different heights from the floor in the aisles between the desks and the outside wall. These temperature readings showed a variation from almost nothing to eight or ten degrees, between the floor and the top of the desks. During the very severe weather of January and February, 1912, it became evident that this installation was only partially satisfactory. When the outside temperature was ten or more degrees below zero, there was a cold current of air that spilled out from the window over the top of the heating pipes above the window sills. This fact led to a proposed installation to overcome the window chill in another way—by means of a sheet of hot air. This new installation will be tried during the coming winter.

The plenum system with which we have worked combines the heating and the ventilating of a building. The heating is accomplished by means of hot air which also is used subsequently for ventilation.

Too much importance cannot be placed on the quality of construction in a building in its relation to the efficiency of the ventilating system. This is particularly true with reference to installation. In the usual type of building construction, it is sometimes necessary, in order to heat the building properly, to introduce more air than is required for ventilation. The other alternative might be to introduce the air at an unduly high temperature, but this procedure is always objectionable because of its effect upon the comfort of the occupants. When separate

means of heating the building are provided, as with direct radiation, there is a tendency to operate the plant without ventilation. In general, the lower it is possible to maintain the temperature of the incoming air, without discomfort, the better are the results with reference to ventilation. From the standpoint of economy, it is always desirable to introduce no greater volume of air than is actually required for ventilating purposes.

Our first attempt at insuring an equitable distribution of air for ventilation purposes within the experimental room led us to a more permanent installation. In the new installation, we have separated the heating of the experimental room from the ventilating, in so far as they seem to impair the efficiency of each other. The scheme in brief consists in bringing the air for ventilating purposes into the room through galvanized iron ducts insulated with asbestos and located under the false floor. This system of ducts terminates in three-inch iron pipes securely fastened to the floor and leading up under the desks. The room is heated by means of hot air driven under the floor. The idea in this scheme is to warm the floor. In order to reduce the effect of window chill, double windows are to be installed, and the heated air from under the floor will be forced upward between the two sets of windows and thus effectually overcome window chill. As in the older installation, the air comes in below the desks and leaves the room through twelve registers in the false ceiling. The air used for ventilating, being introduced through separate insulated ducts, may be at a much lower temperature than that of the air used for heating.

ABSTRACT OF PAPER

This paper is part of a report of the work carried on by the Chicago Ventilating Commission and relates to their experiments in school rooms to check up and determine the adequacy of the standards in use in the Chicago schools.

Experiments to determine the course of air currents in school rooms, experiments to determine the value of taking the air in at a number of openings at the desk line and corresponding outlets at the ceiling.

DISCUSSION

Mr. Edward T. Murphy: Mr. President, in reading this paper over with an idea of discussing this subject, I could put my finger on little to discuss, principally, because, as I understand it, it is in the nature of a progress report, and that even

the new room which they planned for this Fall is not suggested as the entire solution of the problem.

It does seem as if some experiments should be made on economy of operation. For instance, in the suggested means for overcoming the window chill, they abandoned the use of direct radiation, without providing any means for preventing the spilling over of cold air above the pipes. As some of you probably know, in the New York Stock Exchange this was overcome by means of a double sash, with a radiator below an open sill. The inner sash was opened at the top, a current of air dropped to the radiator, and out through a register into the room, thus creating a local recirculation. It might even be possible to locate enough radiation at this point to take care of the entire heating requirements, making it easy to warm the building without the use of any additional air over and above that required for ventilation.

It is also probable that the movement or agitation of the air has something to do with the "comfort zone," although it is not mentioned in their tests so far. A possible arrangement might be the introduction of warm air at the floor through a mushroom designed with an ejector effect, so that a greater volume of air could be moved than that actually introduced by the apparatus.

In an industrial plant, some years ago, we were asked to lay out a system to overcome the usual defect of the hot zone at the top of the room, and a comparatively cool zone at the bottom; that is, to turn the usual room condition upside down. In this installation the air was introduced at the floor line well distributed and also exhausted at the floor line through alternate openings. The effect on the building was 10 degrees warmer condition in the living zone than at the roof line. Of course this was a high building, and this principle might not be applicable to a school room, but it could easily be tried out by exhausting air from the baseboard, introducing it either as proposed for the test room or at the floor line through mushrooms.

Mr. Cooley: This paper recalls to my mind a paper that Mr. Thompson read before this Society about two years ago in regard to the ventilation of the House of Representatives' Chamber at Washington, which a good many of you will recollect. The results there were vastly different from the results that are obtained in this room. In that case, of course, the ceiling was abnormally high. In this case it seemed abnormally low, on ac-

count of the air chambers that they put on the floor and the ceiling, which reduced the height of that room very much.

As I recollect Mr. Thompson's paper, the purest air that was found anywhere was in the exhaust ducts in the ceiling, and it was four to eight times as good as was found around the room in several places, indicating that the air took currents and went up to the ceiling, and there were pockets all through the room which were not ventilated at all. That emphasizes Mr. Murphy's suggestion that the air should be circulated at the same time with the fresh air being brought in; so that there is a circulation of the air in the room to keep those dead pockets down. It is a little questionable whether you would get as good results with a high ceiling as you would with the extremely low ceiling put into that room, certainly in the Capitol they did not get such a result as that.

There is another point here in regard to his comfort zone that suggested itself to me. He has temperature on this chart as low as 55 degrees with 70% humidity, and 68 degrees with 28%, running up as high as 75 degrees.

If you work out the temperature of the wet bulb all the way through, you will find that you have a constant wet bulb temperature throughout the comfort zone. That is an idea I have had a good many years that it was a wet bulb temperature rather than the actual temperature of the room that makes the room comfortable. You will find that that wet bulb temperature, if you work it out on the chart, is practically a straight line across the chart that he has there. It is very interesting to note.

The President: Are there any other remarks on this paper? It will possibly be of some interest to you to know a little of the work that the New York Ventilating Commission is doing along this line. We have found difficulty in using any of the methods suggested here for obtaining a measure of the air movement of the room so that it may be presented in such a way that you can interpret it and understand it days, months or years afterwards. The case involved here is not such that you can take observations over the total area of the room, and put them down in any way, that is, record them.

Mr. Palmer, of our staff, has evolved a rather ingenious method which seems to be working out very well for that purpose. We have two rooms in public school 51, in one of which we have numerous wall inlets and outlets. Then we have another room adjacent, in which we get results similar to those described here, except that for the present we have been using mushrooms under

each seat through which to bring the air in, these being connected with ducts on the ceiling of the room below. In this room we have run ducts on the ceilings with inlets having pans under them, so that the air is distributed and comes down without any noticeable downward current. We have some cranes at the ceiling, so that the apparatus can be moved into any position in the room. From this is suspended a stick, 6 feet long, to which is attached 6 threads, one foot long, at the end of which are one inch square pieces of tissue paper. Now, this is moved over each row of desks and carried the length of the room and the deflection of these strings with the paper on the end thereof is observed. This device can also be raised to different heights, so that there is obtained a survey of the air currents in the room at all levels and at all points. He has also worked out in an ingenious way diagrammatic charts showing the air movements in the room.

Then we took the other room. You can see that with a downward current the strings hanging would not show any horizontal movement of the air, because presumably the air was uniformly moving downward, and its tendency would be to hold the strings straight rather than to move them. So he invented another thing which was a box with glass covered opening in the side and tissue paper hanging across the section thereof, so that a vertical current would at once deflect the paper and indicate the strength of the air movement. It is so sensitive that you could light a match and hold it there and the paper will swing sometimes two inches while holding the match two or three inches below the bottom. This gave a method of measuring the downward movement of the air. When we undertook to do this, however, I was somewhat surprised to find that, according to this device, sensitive as it was, we did not get any movement of air when putting 1,500 cubic feet of air a minute through the room. The amount of air we measured by three methods. We get plenty of air, but at almost no point of the room could we get any movement of air registered on any device. Smoke would flow leisurely around, mostly downward, and balloons would move, but very slowly. That suggests the question whether we are in that room than is desirable. During the Spring of last year the odors in that room were invariably worse than in the other room, whether we used recirculated or outdoor air. In determining odors we used six observers, different teachers selected at different times.

The question is whether we are not getting too good diffusion of air. The air coming into the room at approximately 300 feet velocity through a register undoubtedly does take up more or less of the air of the room, and causes a great deal more of a sweeping motion of the air through the room.

In the matter of the zones of comfort we have made quite an extensive study. We agree very closely with the median here, but the extreme right and left of this chart is rather at divergence with us, being higher at the right and lower at the left.

This air measurement work we have done merely to study technique, and to get some idea of what happens in warm weather, but more as a basis for this winter to determine something of the effects of air currents, and the different circulation methods in the room with different atmospheric conditions outside.

CCCXCI.

FINAL REPORT OF THE PROFESSION'S EFFICIENCY AND WELFARE COMMITTEE*

Pursuant to your wishes expressed at our last Chapter meeting, we are confining our report proper to a short, concise schedule of recommendations, followed by an appendix containing the data and arguments upon which these recommendations are based, together with suggestions as to ways and means of carrying out these and other recommendations.

ITEM NO. I—MEMBERSHIP STANDARDS AND THE LICENSING OF ENGINEERS

It is our best judgment that the Society membership should be a thing to be more sought after, also that a movement should be started for the legal licensing of engineers under the direction of State Boards consisting of engineers selected by the National Society, with perhaps a National Board for deciding appeals from local boards.

ITEM NO. II—THE STATUS OF THE ENGINEER IN HIS OWN FIELD

In this connection we find that the services of the engineer are too seldom employed, and the field we work is not properly cultivated as is shown in the unproductiveness and size of the fees secured.

We also believe that in a great many cases, engineers could be employed to better advantage by the owner direct, as are other professional men. This system would give the engineer more authority for deciding engineering questions, such as is accorded other professional men in the decision of matters pertaining to their work.

There is no working agreement between engineers, architects, contractors and manufacturers, whereby each of these branches

* This report was made by a committee appointed by the New York Chapter. The report was adopted by the Chapter and referred to the Society with a request that it be published in the Society Journal.

might be confined to its own particular field with better economic result to all.

Our recommendations for the correction of these conditions are: —First, that a large portion of the activities of the Society and of its business office be diverted from so many narrow and purely technical matters, and be put to work on the policy and business of engineering. Also that some plan be worked out whereby the engineers can co-operate with this movement in their every-day business life.

Second, that a survey be made and action started toward more general direct employment of engineers.

Third, that a survey be made and action started on a working agreement between engineers, architects, contractors and manufacturers.

ITEM NO. III—ENGINEERING FEES

Your Committee feels that there is a great deal to be accomplished by the standardizing of fees.

We submit the following schedule and recommend a movement toward its adoption by the Society. We also recommend that the work started by the Joint Committee of the A. S. H. V. E. and the A. I. A. be reopened for the purpose of showing the architects the wisdom and propriety of adopting our schedule along with theirs.

	Cost.	Fee.
Residence heating jobs below..	$1,000	$50.00
Repairs and alterations.......		10%
New Work	5,000 or less	10%
New Work	Over $5,000	6%

ITEM NO. IV—THE EFFICIENCY OF ENGINEERING

We feel that there is a great deal of engineering being done poorly and so as to prove uneconomical, simply to save time and cheapen the cost of engineering.

This is at the expense of false economy to clients and a detriment to the profession.

The remedy for this lies in putting engineering on a better business footing and bringing the public to realize more fully that it is a paying investment.

ITEM NO. V—RESPONSIBILITIES OF THE ENGINEERS

The accepted arrangement of affairs to-day undoubtedly places unfair responsibilities upon the engineer. A movement should be started to rectify this condition and to prevent any tendencies toward further unfairness in this particular.

ITEM NO. VI—THE SEPARATION OF CONTRACTS

Generally speaking and on public work in particular, engineering contracts should be separated from other contracts and a movement should be started toward this end.

ITEM NO. VII—CO-OPERATION WITH OTHERS

We should enlist the co-operation of other professions and trade organizations in this work.

ITEM NO. VIII—PRINCIPLES OF PROFESSIONAL PRACTICE

We strongly recommend the formulation and adoption of a code of principles for the guidance of professional practices, modeled somewhat after the codes of other professional bodies.

APPENDIX

The following appendix is offered in the way of elucidation, and sets forth some of the reasons and arguments upon which the above recommendations are based.

The business of engineering like any other business consists of judgment, policy, system, and engineering.

For the most part we are pushing engineering only, and allowing other people to regulate the other important factors of our own business.

Item No. 1: We feel that our Society is not holding the standards of its membership high enough, and are not standing up for the respect and consideration the profession deserves. We would recommend a gradual restricting of the Council's interpretation of the existing By-Laws in this respect with the end in view of revising the By-Laws at a later date, so as to be more specific and restricted.

Item No. 2: There is a great volume of heating and ventilating engineering not being done by heating and ventilating engineers.

To correct this economic defect it is urgently necessary that after qualifying by study, the heating and ventilating engineer take up the equally important business side of his work.

He must ignore the moss covered notion that his services should be sought.

He must see to it that the vàlue of his services are understood and that they are presented with the same frequency and dignified persistency with which the mere mechanical factors in the equipment are offered, explained, pushed, specified, sold and put into actual use.

The sale of engineering services needs to be given a positive impetus.

Engineers are employed for an utterly inadequate proportion of the available work in their field.

During 1914 the cost of building operations in the United States amounted to seven hundred millions of dollars, of which seventy million dollars might be considered the cost of heating and ventilating apparatus and the percentage of this sum which a reasonable fee would bring (say 5 per cent.), has not as yet been discovered by your Committee as having reached our field. It might be considered that of this sum ¾ of a million dollars came to heating and ventilating engineers, leaving 2¾ millions to represent the loss to the profession. Why?. Because the field is not cultivated.

Some of the work not being done by engineers is being done by architects, principally under what we believe to be a false impression, that it can be so handled to the best interests of the work as a whole.

Some by the manufacturers, principally under what we believe to be a false impression, that it is a good and economic business proposition for themselves and the owner.

Some is being passed over, principally under what we believe to be a false impression, that it is not worth doing.

Of the work being done by architects, there is no question but what some of this is being done well and especially is this true where regular engineers are employed on the architects' staff. The fact remains, however, that the guiding genius of an architectural office is not, often, so constituted by nature, that he can look with an impartial eye upon two branches of work so different as architecture and engineering. Self-preservation compels the architectural viewpoint to subordinate the plain points of utility in engineering to the finer points of beauty and arrangement of the building. The result is that the engineering vitals often suffer and have their usefulness curtailed or destroyed; and in a great many cases without the architect's appreciating the gravity of the results.

We believe that the conditions might be improved in many cases by employing a consulting engineer direct and allowing him to be free to act in conjunction with the architect for the best interest of his client.

Of the engineering work being done by manufacturers, some of this is no doubt legitimate and proper for the promotion of new and special devices and systems, and in the furnishing of data upon which manufacturers are sometimes better posted than engineers. A great deal of complaint is heard against engineers for not

being willing to spend the time and trouble to investigate new apparatus, and for not allowing free and open competition. It is argued by some manufacturers that it is necessary that they draw plans and specifications in order to get their share of the business. It is also argued that engineers are sometimes prone to undertake work for which they are not competent, and for which they lack the necessary experience. That they are further prone in a great many instances to spurn any data or consultation from the manufacturers. Under such conditions the manufacturer sometimes argues that it is a matter of self-defence for him to draw plans and specifications. We feel that in a great many cases the engineer is in the wrong and should as speedily as possible put himself in the right. Where these conditions do not exist, however, we feel that the system of manufacturers' drawing plans and specifications is wrong both from an ethical and an economical standpoint.

In the first place, the manufacturer is generally forced to draw plans and specifications for no other compensation than what can be added to the cost of the work. This introduces the greatest temptation to use special apparatus that no one else can furnish, whether the same is suitable for the purpose or not. He is also forced, in order to include his own apparatus, to lay out the entire plan, including piping, electric and transmission systems, with which his engineering staff is naturally not so familiar as with his own apparatus and its immediate connections. In other words, the greater part of the layout is handled as a side issue and is not so completely handled as it would be by a regular engineer making such matters his whole business. He is also generally thrown into competition with other contractors, who are drawing their own plans and specifications, and as a result there are a number of such, all costing their respective concerns almost as much as one good set would cost a regular engineer. All of these costs must be borne sooner or later by the consumer.

We feel that a thorough understanding and a square deal by all parties concerned under this arrangement would work toward its immediate abatement and the final elimination of these conditions.

We find that the Government, and a great many of the Municipal Departments throughout the country are maintaining either separate or co-ordinate engineering and architectural departments, and that the State Commissions and other similar bodies are requiring that architects in competition shall employ engineers satisfactory to the Commission. It is evident that in such cases the value of the proper handling of the engineering work is being appreciated. It is also noted that among the contractors there are quite a few now who

are refusing to bid upon engineering plans and specifications which are not directly in charge of an engineer. This movement we understand is not prompted by any personal or political motives, but is born of a spirit of self-preservation of the financial and engineering reputation of such concerns, against the making good of unforseen troubles and the changing of work in place, where details are not properly worked out, and other evil effects of unsatisfactory engineering.

As to what is being done toward the elimination of the drawing of plans and specifications by manufacturers, some agitation has been carried on by this Society, and the manufacturers themselves have had the matter up for serious consideration since 1910. As a result, a great many of the manufacturers have discontinued the practice, and the majority of the rest, we believe, are willing to do so as soon as they can see that it is to their advantage.

Item No. 3: We feel that there is a great deal of engineering work being done for less compensation than will provide for its being done properly. A great many engineers appear to be sacrificing their own reputations, the high standards of the profession and the interest of the work as a whole, to commercialism and their own immediate interest; in other words, there seems to be a tendency to strive for large business at a small fee, and to turn out work which will not do credit to the profession. We feel that we should get away from this department store way of looking at engineering.

An attempt to formulate some schedule of fees discloses the fact that in 1900 and 1901 there was a conference between committees representing our Society and the American Institute of Architects that resulted in an agreement that the owner should pay for such engineering services as the architect required. It remained for the Society to formulate a schedule of fees for engineering services, and to reach some agreement regarding same with the architects, but as far as can be discovered this was never done.

We would call your attention to the following provisions which now exist in Section 3 of the American Institute of Architects' Principles of Professional Practice:

"ON SUPERINTENDENCE AND EXPERT SERVICES."

"On all work except the simplest, it is to the interest of the owner to employ a superintendent or clerk of the works. *In many engineering problems and in certain specialized esthetic problems, it is to his interest to have the services of special experts and the architect should so inform him. The experience*

and special knowledge of the architect make it to the advantage of the owner that these persons, although paid by the owner, should be selected by the architect under whose direction they are to work."

We consider these provisions indefinite and subject to improper use and interpretation. As such they form one of the weak points in what is otherwise a very rigid set of safeguards which the architects have chosen for regulating their professional ethics.

We believe that the architects could be shown that it would not only be to our advantage, but to theirs as well, to amend this section so as to include a minimum schedule of fees for engineers, which would eliminate inferior engineering and its reflection upon our profession and theirs.

As the matter now stands, the architect in many cases collects the whole fee and pays the engineer out of same. With this arrangement the architect who would not break the rules of his profession by cutting his own fee may accomplish the same end by cutting the engineer's fee with impunity. This means that the architect who uses and pays for good engineering service, and by so doing is serving both his profession and ours and the public honestly, is at an unfair disadvantage with the architect who will cut the fees for the engineering.

The most important ethical point for the Society to cover in the matter of fees is that in reference to residence in which the service of the engineer is valuable to the last degree.

For such work the engineer is so seldom employed that charges cannot be reliably given.

Some engineers hold that the minimum should be $50.00; others say 10 to 15 per cent. and they claim to do some work at this rate. Others say not less than 6 per cent.

Others claim that for factory, school, church and public buildings, the work be divided into direct radiation plants and blower plants.

Some office costs which have come to our attention run from 3½ to 4 per cent., including inspection.

The above figures are for average size jobs and will vary some, but are considered a fair average and sufficient to show that practically all of the engineering which is being done below 5 per cent. is either being done at a loss, or is not productive of work which will probably be satisfactory to clients or creditable to the profession.

We have nothing now for the man who wants to obey the rules of law and order to conform to, nor for any engineer to point to as the recognized value of his services. Accordingly we have worked out a schedule of minimum charges.

A sliding scale schedule has been selected for three reasons: First, because other professional men have found it necessary and proper; second, because the office costs do not vary directly with the cost of the work; and third, because engineers will naturally scale their prices down as the size of the job increases, rules or no rules.

Item No. 4: We also find indications tending to show that engineers are not turning out the best work that is in them and are not drawing the line closely enough between good work and short cut methods. As an illustration, the greatest number of piping, duct and transmission systems are designated from average tables, without going to the trouble to work out figures for particular cases.

Such individual handling of particular cases would often result in substantial savings from lower cost and better operation as a return for a comparatively small extra engineering fee. Such matters must be explained to the owners and architects, however, and we find our own members reluctant in using such means in promoting the advancement of the profession.

Item No. 5: We also find an indication of what we believe to be a tendency on the part of engineers to assume more responsibility, and to allow more of the burden of proof to be thrust upon them than is true of other professions.

We also find owners and architects demanding that engineers guarantee and bond the proper working of their plants. Also a few engineers who are siding with this view. At the same time the architects and other professional men are not guaranteeing the results of their services, but are simply striving for the best and allowing the client to take the risks. There is nothing in the ordinary engineering fee to cover more than the barest compensation for services rendered, and surely nothing at all to cover any insurance of the risk, which is naturally attached to every business venture.

If such an extra fee is charged and understood the conditions are, of course, different, but it seems to us that this favoring of the engineers' guaranteeing their work is simply placing upon them an unfair handicap not borne by any other profession.

In this same connection we note a tendency on the part of engineers to allow architects and others to shift the whole responsibility for the fitting of the engineering equipment into place (with the building and other items), on to the engineer's shoulders. The engineer is expected, not only to furnish all data as to sizes and requirements of his apparatus (all of which he should furnish, of course), but is sometimes required to follow up and check architectural plans and see that these requirements are met. Furthermore

he is expected to check the architectural plans against all interference between his work and that of others. After this he is required to follow up on the job and see that the architect's plans and specifications are followed out, as far as his work is involved, and that nothing is installed to interfere with the engineering work.

This involves looking after a great volume of work upon which no fee is ever figured for the engineer. He is also expected to be perfectly familiar with the building and all other kinds of plans, whereas, others on the job are most generally very ready to disclaim any knowledge of the first meaning of an engineering plan.

Furthermore, we feel that too many engineers are apt to allow their contractors to be burdened with unfair responsibilities and burdens of proof. Such as the changing of work in place, where the fault lies with someone else, and the sole proving of their claims in this connection, whereas, some of the burden should rest with the other contractors involved. The unfair treatment of a contractor in this way tends to increase the cost of the next job and to embitter the contractor against engineers.

We also note, under this heading of the shifting of responsibility, the tendency of engineers to shift a part of their own responsibilities, such as supervision and details, to others, so as to cheapen their services; but, of course, this is at a greater cost to the job.

Item No. 6: We believe that a great deal of harm is resulting from the practice of letting the engineering contracts as a part of general contracts, if the general contractor has neither financial nor moral responsibilities. In a number of States there is a law against this on public work.

The Master Fitters and Plumbers are working in the interest of such laws in a number of States. In other localities the engineering contractors are handling the situation by refusing to bid to general contractors of the kind mentioned.

Also at the last meeting of the American Institute of Architects, in New Orleans, the following resolution was passed:

> "Resolved, that the American Institute of Architects, in convention assembled, recommends to the members of our profession the adoption of the practice of direct letting of contracts for mechanical equipments, such as heating apparatus, plumbing and electrical equipment.
>
> This resolution is based on the conviction that direct letting of contracts, as compared with sub-letting, through general contractors, affords the architect more certain selection of competent contractors and more efficient control of execution of work, and thereby insures a higher standard of work, and,

at the same time, serves more equitably the financial interest of both the owner and contractor."

Following this, in July, 1914, the National Electrical Contractors' Association passed the following resolution:

"Resolved, that the National Electrical Contractors' Association of the United States, in convention assembled, concurs in the resolution adopted by the American Institute of Architects last December, at New Orleans, covering the segregation of plumbing, heating and electrical equipment, from building contracts; and that a copy of this resolution be sent to the Secretary of the American Institute of Architects."

The proper handling of such work involves the maintenance of departments; they are not as capable of properly supporting as are the engineers, who are regularly engaged in this particular work.

They are also naturally lacking in the proper viewpoints which are very essential to the best interests of any undertaking.

This does not mean, of course, that all work so handled is suffering, but it seems to be pretty well proven, that the system is objectionable and that abuses are being practiced to such an extent as to warrant the serious consideration by all parties concerned, of changing the system, especially on all public work.

The following might be cited as examples of some of the abuses, which are working to the detriment of the engineer and subcontractor.

We have all realized for some time, that the irresponsible general contractor, in a great many cases, does most all of his business on the sub-contractor's capital as follows:

The general contractor, receiving the payments, naturally holds the whip hand, and upon one pretext or another, may hold up payments, thus accumulating money upon which to run his business, while the sub-contractor must meet his bills for material and labor in the meantime. If the general contractor fails, the sub-contractor has no recourse on the owner, who can take his labor and material for which he has received no pay, just the same as that for which he has been paid.

The sub-contractors are also saddled with all kinds of claims for delays and penalties and with the responsibilities for noncompliance with plans and specifications, and in a great many cases very unreasonably so, and may be made to submit under penalty of not receiving their payments.

It is also difficult to secure an equitable settlement of disputes and claims arising between the work of the general contractor and that of the sub-contractors, due to the fact that the other parties con-

cerned are represented before the architect and owner by the general contractor, who is naturally their opponent in such matters.

Again the owner may pay the general contractor a fair price for his equipment (based upon a good fair price from the sub-contractor to the general contractor at the time bids are taken) and the general contractor may shop around and either beat his sub-contractor down, or get an incompetent cheap substitute to do the work. Of course, in either case, the work is never as well done by an incompetent man, or by a competent one who has been beaten down in price, as by a competent one at a fair price.

Again, if the principal representative (among the contractors) is one having the greatest interest at heart in the building only, he will make the equipment, which is of great importance, so subservient of the rest, that it is impossible for the engineer or sub-contractor to secure the best results.

Also in cases where contracts are taken too low, or where the contractor may be desirous of securing an unfair profit at the expense of the owner, it is an easy matter for him to squeeze this out of sub-contractors, upon whom the blame can be laid and against whom the owner has no recourse. In the case where a general contractor takes work too low, he generally fails to realize this until most of the building contracts are let, and then whatever deficiency is to be recouped, must be made up out of the equipment.

In all of the above arrangements, the sub-contractor is the first loser, but the architect, the engineer, the general contractor, and the owner are perhaps greater losers, in the quality of the work, and general results attained; since the sub-contractor could well afford to do work better and cheaper under fairer and more agreeable conditions.

The principal argument the owner and the architect have against any of the above, is that by the employment of a general contractor, they are relieved of a great deal of responsibility and have only the one head to deal with, who is responsible for the full completion and co-ordination of the several branches of the work.

This argument can be shown to be a small matter, however, in comparison with the opposing evils as outlined above. It is a further fact, too, that the architect and owner by not coming into direct contact with the equipment work, have no direct assurance of what they are getting, or that this very important branch of the work is getting a fair and proper consideration. This whole matter goes back to the old proverb that "Whatever is worth doing, is worth doing right," which means that the architect and owner will always be well paid for their trouble in this connection.

As to disputes which may arise among several separate contractors, these arise just the same under general contract. The only difference is, that the architect and owner do not hear so much about them, as a consequence, the work suffers.

These differences practically disappear whenever each contractor fully understands that the owner is to be the final court of decision and that each will get a fair and equitable hearing.

The above outlines in a way the work which we consider to be before the Society on these matters, and we are submitting below our recommendations as to how to proceed with the work.

METHOD OF PROCEDURE

First. We would counsel deliberation and a satisfaction with small returns to start with.

Second. We would recommend the inviting of the appointment of a committee of one (1) from the architects, owners, contractors and manufacturers, to co-operate with us in this movement. It should be made plain to these outside bodies that they are not to be committed to anything by the action of their representative with us, but that they will be thus kept in touch with the movement and would be expected to act through their own body as they saw fit.

Third. We would recommend circularizing the engineers of the East, sending each a copy of this report with the following questions:

> Just indicate on a plain piece of paper the amount of your last year's fees and the highest, lowest, and average percentage which you received for your work, and mail same in a plain envelope to the Chairman of this Committee.
>
> Also please send us your comments on the correctness and value of this movement, and state as to whether you will give same your support.

Fourth. We would recommend circularizing the engineering contractors throughout the East with a copy of this report and the following questions:

> What is your opinion as to the correctness and value of this movement?
>
> Will you give same your support and co-operation?
>
> State the volume in dollars and cents of your last year's business.
>
> What percentage of this was handled by engineers who are not contractors?

What percentage by engineering contractors?

What percentage by architects without regular engineering services?

What percentage without any engineers?

State what you know about the relative satisfaction to all concerned as given by the different jobs under the above differing methods of handling the engineering?

Your reply may be on plain paper, unsigned if desired, but please reply to the best of your ability.

Fifth. We would recommend that the above data be used to start a set of statistics on the causes and effects of our status and the devising of means to better same. Also that this be followed out in the continued gathering of statistics and data to guide us in this work.

Sixth. We would call your particular attention to the fact that practitioners in other professions require to be licensed by law and to have a State Certificate before they are allowed to practice. This is not true of the engineer and some of the above conditions are the result. We would recommend a campaign for the passage of laws on this subject in the various States.

Seventh. We would recommend further that immediate steps be taken to interest every one possible in this work and that everyone of our members make it a special point to so interest himself. We would also recommend and most earnestly request that every engineer first square himself with this movement and offer his objections and recommendations, and then after it is turned out as our finished product, not of this Committee, but of all of us, let everyone constitute himself a committee of one to promote the idea. We do not wish to be misunderstood, or to see any misunderstanding creep in which would stamp this as a revolutionary movement designed to create any Utopian conditions in the immediate future. We do believe, however, that there is a lot to be gotten out of concerted action in this field, and that a continuation of this work will produce results which will accumulate for the good of the profession.

We would suggest that individual efforts be not made in a spirit of martyrdom, or be left to the over enthusiasm of a few and the selfishness of many. Rather let each one enter on the more lasting basis of not sacrificing any more to the cause than he can continue to contribute, but upon this basis let everyone stick to it.

In other words, let everyone throw his force in the right direction and then be his own judge as to where to bend in the meeting of every-day business conditions, but to always be straightening his path toward the goal.

Respectfully submitted,

(Signed) PERRY WEST, Chairman,
FRANK K. CHEW,
J. I. LYLE.

In Memoriam

	Joined the Society	Died
L. H. Hart, New York	Sept. 1894	Jan. 26, 1897
James W. Gifford, Attleboro, Mass.	Jan. 1898	July 26, 1899
William McMannis, New York	Sept. 1894	Jan. 19, 1901
Charles F. Tay, San Francisco, Cal.	Jan. 1896	Sept. 8, 1901
Arthur H. Fowler, Philadelphia, Pa.	Jan. 1897	June 3, 1903
Stephen G. Clark, New York	Dec. 1902	Feb. 3, 1904
Charles M. Wilkes, Chicago, Ill.	Jan. 1897	Jan. 7, 1905
James Curran, New York	Dec. 1901	Oct. 27, 1905
Herbert W. Nowell, New York	June 1904	Mar. 25, 1905
Enoch Rutzler, New York	July 1901	Feb. 29, 1908
Harry J. Ott, Chicago, Ill.	Dec. 1906	Sept. 25, 1908
Thomas J. Waters, Chicago, Ill.	Sept. 1894	Feb. 25, 1909
Max J. Mulhall, New York	June 1909	July 30, 1909
Walter B. Pelton, Dorchester, Mass.	June 1910	Nov. 2, 1910
R. Barnard Talcott, Denver, Colo.	June 1899	Dec. 4, 1910
William H. Bryan, St. Louis, Mo.	July 1898	Dec. 8, 1910
James R. Wade, St. Louis, Mo.	Dec. 1909	Mar. 9, 1911
James Mackay, Chicago, Ill.	Sept. 1894	July 18, 1911
Warren S. Johnson, Milwaukee, Wis.	Jan. 1906	Dec. 5, 1911
W. C. Bryant, Holton, Kan.	Jan. 1901	April 6, 1912
H. A. Joslin, Boston, Mass.	Jan. 1896	Oct. 3, 1912
Andrew Harvey, Detroit, Mich.	Jan. 1896	Oct. 9, 1912
N. P. Andrus, Brooklyn, N. Y.	Sept. 1894	Jan. 13, 1913
J. A. Payne, Jersey City, N. J.	Sept. 1894	Mar. 3, 1913
J. S. Billings, New York	Jan. 1896	Mar. 10, 1913
Wiltsie F. Wolfe, Philadelphia, Pa.	Sept. 1894	Dec. 4, 1913
R. C. Clarkson, Philadelphia, Pa.	Sept. 1894	Dec. 26, 1913
H. W. E. Muellenbach, Hamburg, Ger.	July 1903	April 17, 1914
W. A. Gates, Oklahoma City, Okla.	July 1907	May 24, 1914
F. L. Busey, Buffalo, N. Y.	Sept. 1911	June 7, 1914
E. B. Denny, Newark, N. J.	Jan. 1906	July 12, 1914
A. B. Franklin, Boston, Mass.	June 1902	Aug. 22, 1914

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